# Compressed Air

A MONTHLY MAGAZINE DEVOTED TO THE USEFUL APPLICATIONS OF COMPRESSED AIR.

VOL. XI. NEW YORK, MARCH, 1906. No. 1.

"Compressed Air,"

OR

THE COMPRESSED AIR MAGAZINE.

A MONTHLY PUBLICATION DEVOTED TO THE USEFUL APPLICATIONS OF COMPRESSED AIR.

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VOL. XI. MARCH, 1906. NO. 1

## A New Volume.

With this issue COMPRESSED AIR begins its eleventh year. We have in prospect for the near future several changes in the appearance and arrangement of the magazine which will, we hope, serve to make it more attractive to our readers.

COMPRESSED AIR differs essentially from most of the popular or technical magazines in that it deals exclusively with one subject and stands alone in that field. From the time of its conception it has aimed to sum up the current information on the subject of compressed air and to assist in the discussion and publication of such ideas and suggestions as may deal with its development.

While compressed air was by no means unknown a decade ago, this period has shown a remarkable change in the general attitude of both the general and engineering public toward this method of power transmission. In wide and distinctive fields compressed air has found its place. There are still many opportunities before it which time will develop.

COMPRESSED AIR is not the organ of an experimental science but deals with practical subjects. The new inventions and the experiments all have their place in our columns but it has been our consistent policy to deal at length with the practical uses of this power and to include much that can be read with interest and profit by the man who is called upon to use air under pressure in the course of his daily work.

Descriptive technical articles are published in these columns at frequent intervals. One of special importance is given space in this issue. It deals with the different formulae relating to the compression of air and should be preserved by every engineer who has to deal with air under pressure.

Ten years ago it was an open question whether there was a sufficiently large reading public among those interested in compressed air to support a publication devoted exclusively to that subject. COMPRESSED AIR was started and the course of events has conclusively proven that it has its place. The growth has been a steady and a healthy one. The same policy in the conducting of this magazine which has been successful in the past will be followed in the year to come. Effort will be made to increase its usefulness in its particular field. In this we ask the co-operation of our readers.

Suggestions as to ways in which the magazine can be improved, will be cordially welcome. Its columns are always open to contributions on any feature of the general topic. If you have problems in the use of compressed air regarding which you desire assistance, write to us. We will gladly help you or put you in touch with some one who can. Such inquiries often prove of general interest and start a discussion which will be of material assistance to many.

We thank our advertisers for their support during the last twelve months. COMPRESSED AIR does not claim a circulation equal to many of that of the general engineering publications. It has, however, this one advantage: Every copy goes to some one who is interested in compressed air. If you would reach the people who are looking for information on this subject, COMPRESSED AIR offers the only practical medium.

## A Compressed Air Saw.

Readers of COMPRESSED AIR may remember a description of the Redfield compressed air saw which appeared in the February, 1904, issue. We have had a number of inquiries for that device since that article appeared, the latest coming from a man interested in a northern Minnesota timber proposition. He contemplates beginning work at an early date and wants to use the very latest improved appliances. Having heard of this saw in question, he wrote to us for information. The saw is built by the Ashland Iron Works of Ashland, Oregon, and was given its first practical trial in the woods by the McCloud River Lumber Company, of McCloud, California.

If any readers of COMPRESSED AIR have any further information regarding either this saw or any other operated by compressed air, we shall be pleased to give it space in the coming issues of this publication. There appears to be a distinct field for devices of this nature for use in the western logging camps.

## Sea Level or Lock Canal.

Without doubt, the greatest engineering project of the present century is the task which the United States Government has assumed in the building of the Panama Canal. Included in this is the removal of over 100,000,000 cubic yards of rock. In this gigantic undertaking compressed air is bound to come to the fore as an absolute essential for the rapid and economical completion of this part of the work. For that reason readers of COMPRESSED AIR will find it of interest to keep in touch with the progress being made.

The last step has been the forwarding by President Roosevelt to Congress of the report of the Board of Consulting Engineers. This report consists of two parts, one being the plan of eight members of the Board and in favor of the sea-level canal; the other being the report of five members of the Board recommending the canal with a summit of eighty-five feet above the sea, to be reached by locks. The report was accompanied by a letter to Secretary of War Taft from Chief Engineer Stevens and the members of the Isthmian Canal Commission, with the exception of Admiral M. T. Endicott. The members of the Commission and Mr. Stevens endorsed the minority report, which favors the eighty-five foot lock canal. Admiral Endicott advocated the sea-level structure, as outlined in the majority report. President Roosevelt's letter of transmissal places him among those who favor the lock canal.

A careful perusal of the report makes it clear that the majority has based its recommendation on the belief that the greater efficiency and ultimate value of the sea-level canal make it more than worth the extra cost and the slightly increased time of construction it requires. Its international character makes the conditions under which it will be operated so radically different from those of similar enterprises on a smaller scale that it is hardly fair to judge one by the other. While the completion of the canal at as early a date as possible is a thing much

to be desired, the question of a few years should not be allowed to interfere and impair the efficiency of an undertaking of this nature which is destined to serve the people of this and other countries for many generations to come.

Every citizen of the United States, particularly those who are interested in the engineering projects of the land, should not fail to become conversant with the plans of construction now under consideration. Through the courtesy of the *Engineering Record,* COMPRESSED AIR will give in a coming issue an illustrated resume of the essential features of the sea-level canal as advocated by the majority of the Board of Consulting Engineers. An abstract of the plan for the lock canal as proposed by the minority will be given later.

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## Derivation of Formulae for Single and Stage Compression; also, Proof of Conditions Governing Best Proportioning and Highest Economy in Stage Compression.

BY EDWARD F. SCHAEFER, M. M. E.

In estimating various items in air compression, the usual and, of course, the natural and desirable way of proceeding is to employ formulae which have previously been worked out. This practice saves time, and eliminates, to a large extent, errors that would be apt to appear in long calculations.

Of all the equations used in solving problems, there is hardly one more important or more frequently employed than that which expresses the *work done* in compressing air or gas. It is the author's purpose to show how this useful formula for horse-power is developed.

Adiabatic compression occurs when a decrease in volume is attended by an increase in both pressure and temperature, according to the law:

$$p_1 v_1^{1.41} = p_2 v_2^{1.41} = \text{Constant}$$

In this equation $p_1$ and $v_1$ are the pressure in pounds per square foot and the volume in cubic feet, respectively, at a certain point during compression or expansion; $p_2$ and $v_2$ are the pressure and volume at another point.

Although in practice pure adiabatic compression is impossible—nor is it desired—the actual compression approximates the adiabatic closely. We assume, therefore, that the two are identical; in any case any slight error is on the safe side.

In stage compression the same assumption holds, but to reduce the work done on the air and increase the efficiency of the compressor the air is cooled between stages by means of an intercooler.

Fig. 1 shows an ideal indicator card, EG represents the vacuum and CH the atmospheric lines. Beginning at H, we find that air is sucked into the cylinder to C and at atmospheric pressure $p_1$; at C compression begins—this compression is adiabatic and continues to A, a point where the exhaust valves open; along AF and at the pressure $p_2$ air is transferred from the compressor cylinder to a receiver or intercooler, as the case may be. At F the admission valves open, the pressure in the cylinder falls to $p_1$ and a fresh charge of air is taken into the cylinder. Thus the cycle is represented by the figure HCAF. This area is equivalent to the work done in compressing and transferring a cylinderful of air.

Work done = ACEGF — HCEG
or work done = ADEC + AFGD — HCEG
But AFGD = $p_2 v_2$
and HCEG = $p_1 v_1$
$\therefore$ work done = ADEC + $p_2 v_2 - p_1 v_1$ ..............................(1)

It remains now to show what the value of the area ADEC is in terms of $p_1$, $v_1$, $p_2$ and $v_2$.

From Calculus we know that:

$$\text{ADEC} = \int p \, . \, dv \quad \ldots\ldots\ldots\ldots\ldots\ldots (2)$$

in which $p$ represents any pressure from C to A, and $dv$ any increment in volume along ED.

The law governing adiabatic compression is:

$$p v^n = p_1 v_1^n \quad \ldots\ldots\ldots\ldots\ldots\ldots (3)$$

in which the index $n$ is equal to 1.41.

Re-writing (3) we get:

$$p = \frac{{}_1 v_1^n}{v^n} \quad \ldots\ldots\ldots\ldots\ldots\ldots (4)$$

In compressing air from C to A, the volume changes from $v_1$ to $v_2$, and therefore:

$$\text{ADEC} = \int_{v_2}^{v_1} \frac{p_1 v_1^n}{v^n}\, dv \ldots\ldots\ldots\ldots\ldots (5)$$

$$= p_1 v_1^n \int_{v_2}^{v_1} \frac{dv}{v^n}$$

$$= p_1 v_1^n \int_{v_2}^{v_1} v^{-n}\, dv$$

$$= p_1 v_1^n \left(\frac{v^{-n+1}}{-n+1}\right)_{v_2}^{v_1}$$

Reducing this expression to a common denominator, simplifying the numerator, reducing and remembering that $p_1 v_1 = p_2 v_2$, we get:

$$\text{Work done} = \frac{n}{n-1}(p_2 v_2 - p_1 v_1)$$

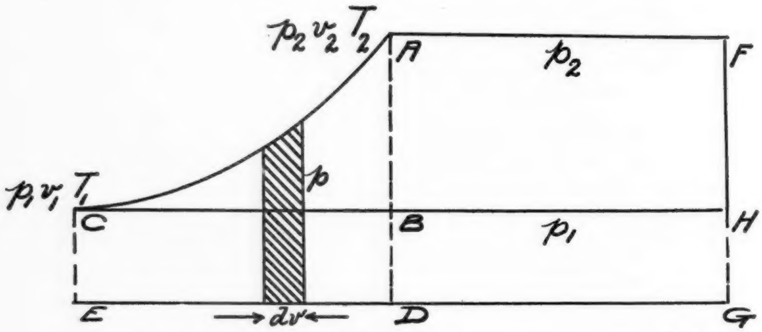
Substituting for $n$ its value 1.41, the following is deduced:

$$\text{Work done} = 3.45\,(p_2 v_2 - p_1 v_1)$$

$$\text{But } p_2 = \frac{p_1 v_1^n}{v_2^n}$$

$$\text{Work done} = 3.45\left(\frac{p_1 v_1^n v_2}{v_2^n} - p_1 v_1\right)$$

$$= 3.45\, p_1 v_1 \left(\frac{v_1^{n-1}}{v_2^{n-1}} - 1\right)$$




2203 Fig. 1 E.F.S

$$= p_1 v^n (v_1^{-n+1} - v_2^{-n+1}) \div (-n+1)$$

$$= \frac{p_1 v_1^n}{n+1}(v_2^{1-n} - v_1^{1-n})$$

Multiply by $(v_1^{1-n}) \div (v_1^{1-n})$

$$\text{ADEC} = \frac{p_1 v_1}{n-1}\left(\frac{v_2^{1-n}}{v_1^{1-n}} - 1\right)$$

$$= \frac{p_1 v_1}{n-1}\left(\frac{v_1^{n-1}}{v_2^{n-1}} - 1\right)$$

Substituting this result in equation (1) we have:

$$\text{Work done} = \frac{p_1 v_1}{n-1}\left(\frac{v_1^{n-1}}{v_2^{n-1}} - 1\right) + p_2 v_2 - p_1 v_1$$

$$= \frac{p_1 v_1 v_1^{n-1} - p_1 v_1 v_2^{n-1}}{(n-1)\, v_2^{n-1}} + p_2 v_2 - p_1 v_1$$

From equation (3) we have:

$$\left(\frac{v_1}{v_2}\right)^n = \frac{p_2}{p_1}$$

$$\text{Or } \left(\frac{v_1}{v_2}\right)^{n-1} = \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}$$

Substituting the result just found in the above equation, we have:

$$\text{Work done} = 3.45\, p_1 v_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} - 1\right]$$

And this expression, when reduced by using the value of $n$, becomes:

$$\text{Work done} = 3.45\, p_1 v_1 \left[\left(\frac{p_2}{p_1}\right)^{0.29} - 1\right] \ldots (6)$$

This equation expresses the amount of work done in foot pounds to compress and deliver one cubic foot of free air, adiabatically, from a pressure $p_1$ to another pressure $p_2$.

By dividing by $v_1$ the expression will represent the mean effective pressure (M. E. P.) per stroke, or,

$$\text{M. E. P.} = 3\ 45\, p_1\left[\left(\frac{p_2}{p_1}\right)^{0.29} - 1\right] \ldots\ldots (7)$$

and therefore

$$\text{H. P.} = \frac{144}{33000}.\ \text{M. E. P.}$$
$$= .00436\ \text{M. E. P.} \ldots\ldots\ldots\ldots\ldots (8)$$

in which the pressures are reduced to pounds per square inch.

*Example I.*

Required the horse-power necessary to compress one cubic foot of free air adiabatically to 90 pounds per square inch gauge pressure in a single-stage.

$$\text{M. E. P.} = 3.45\, p_1\left[\left(\frac{p_2}{p_1}\right)^{0.29} - 1\right] \ldots\ldots (7)$$

$p_1 = 14.75$ pounds per square inch
$p_2 = 14.75 + 90 = 104.75$ pounds per square inch

M. E. P. $= 3.45 \times 14.75\ (1.76 - 1) = 38.68$

or H. P. $= .00436 \times$ M. E. P. $= 0.169$ per cubic foot of free air in *single stage* compression.

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It was proved by F. A. Halsey that in proportioning compound compressor cylinders, best results are obtained by dividing the work equally between the cylinders, or by proportioning them so that maximum saving in power is effected (*American Machinist,* March 31, 1898). Letting

$$\frac{p_3}{p_1} = \text{R}$$
$$\frac{p_2}{p_1} = r_1$$
$$\frac{p_3}{p_2} = r_2$$

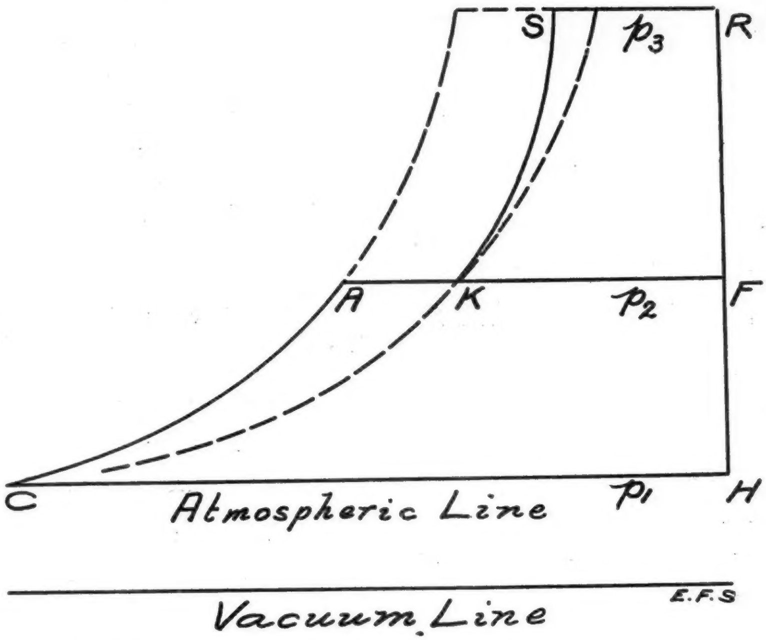


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Fig. 2

Mr. Halsey found that for best proportioning

$$r_1 = r_2 = \sqrt{R} \ldots\ldots\ldots\ldots\ldots\ldots (9)$$

Now $\frac{p_2}{p_1} \times \frac{p_3}{p_2} = r_1 \, r_2 = R$

$$\therefore \; r_2 = \frac{R}{r_1}$$

If $W_{l.p.}$ = work in low pressure cylinder
$W_{h.p.}$ = work in high pressure cylinder

then from (6)

$$W_{l.p.} = 3.45 \, p_1 \, v_1 \, (r_1^{0.29} - 1) \ldots\ldots (10)$$

$$W_{h.p.} = 3.45 \, p_2 \, v_2 \, (r_2^{0.29} - 1) \ldots\ldots (11)$$

The total work done by the compressor is equal to the sum of the work done in the low pressure and that done in the high pressure cylinder, or,

$$(W_{l.p.} + W_{h.p.})$$

$$W_{l.p.} + W_{h.p.} = 3.45 \, p_1 \, v_1 \, (r_1^{0.29} - 1)$$
$$+ 3.45 \, p_2 \, v_2 \, (r_2^{0.29} - 1) \ldots\ldots (12$$
$$p_1 v_1 = p_2 v_2$$
$$= 3.45 \, p_1 \, v_1 \, (r_1^{0.29} - 1 + r_2^{0.29} - 1)$$
$$= 3.45 \, p_1 \, v_1 \, (R^{\frac{0.29}{2}} + R^{\frac{0.29}{2}} - 2)$$
$$= 3.45 \, p_1 \, v_1 \, (2 \, R^{0.145} - 2)$$
$$= 6.90 \, p_1 \, v_1 \, (R^{0.145} - 1) \ldots\ldots (13)$$

The equation expresses the amount of work done in foot pounds to compress and deliver one cubic foot of free air, adiabatically, from a pressure $p_1$ to another pressure $p_3$ in *two* stages.

Dividing (13) by $v_1$ we get the M. E. P. reduced to the low pressure cylinder.

$$\text{M. E. P.} = 6.90 \, p_1 \, (R^{0.145} - 1) \ldots\ldots (14)$$

$$\text{H. P.} = \frac{144}{33000} \text{M. E. P.} = .00436 \text{ M. E. P.} (15)$$

in which the pressures are expressed in pounds per square inch.

*Example II.*

Required the horse-power necessary to compress one cubic foot of free air adiabatically to 90 pounds per square inch gauge pressure in two-stage compression.

Fig. 2 shows the ideal combined cards from a compound compressor.

As stated above, after compressing the air adiabatically from C to A the heat due to the compression is removed by passing the air through an intercooler. The temperature falls to that of the free air and the volume decreases from A to K. K is the point where the isothermal line cuts the line representing the receiver pressure.

In equation (13) $R = \frac{p_3}{p_1}$

$p_1$ = 14.75 pounds per square inch.
$p_3$ = 14.75 + 90 = 104.75 pounds per square inch.

$$\text{M. E. P.} = 6.90 \times 14.75 \, [(7.12)^{0.145} - 1]$$
$$= 6.90 \times 14.75 \, [1.329 - 1] = 33.48$$

or H. P. = .00436 × M. E. P. = 0.145 per cubic foot of free air in *two-stage* compression.

In general, therefore, horse-power, whether *single-* or *two-*stage, is expressed by

$$\text{H. P.} = .00436 \text{ M. E. P.}$$

For single-stage compression:

$$\text{M. E. P.} = 3.45 \, p_1 \left[ \left( \frac{p_2}{p_1} \right)^{0.29} - 1 \right]$$

For two-stage compression:

$$\text{M. E. P.} = 6.90 \, p_1 \left[ \left( \frac{p_3}{p_1} \right)^{0.145} - 1 \right]$$

These three equations will be found adequate for general compressor operations and are, as here given, in the simplest possible forms.

At times, however, circumstances are such that triple or quadruple compression is necessary, and for such cases the formulae given below are to be employed.

For three-stage compression:

$$\text{M. E. P.} = 10.35 \, p_1 \left[ \left( \frac{p_3}{p_1} \right)^{0.097} - 1 \right]$$

For four-stage compression:

$$\text{M. E. P.} = 13.80 \, p_1 \left[ \left( \frac{p_3}{p_1} \right)^{0.0725} - 1 \right]$$

For five-stage compression:

$$\text{M. E. P.} = 17.25 \, p_1 \left[ \left( \frac{p_3}{p_1} \right)^{0.058} - 1 \right]$$

In general:

$$\text{M. E. P.} = 3.45 \, N p_1 \left[ \left( \frac{p_3}{p_1} \right)^{\frac{0.29}{N}} - 1 \right]$$

in which N represents the number of stages in which the compression is to be accomplished, $p_1$ the intake absolute pressure and $p_3$ the terminal absolute pressure.

Occasions frequently arise when it is of importance to be able to determine with dispatch the horse-powers needed to compress air, having given the terminal pressure.

In the formulae given the only quantity necessary is the terminal absolute pressure.

For single-stage compression:

$$\text{H. P.} = 0.10163\,(p^{0.29} - 2.183)$$

For two-stage compression:

$$\text{H. P.} = 0.30043\,(p^{0.145} - 1.477)$$

For three-stage compression:

$$\text{H. P.} = 0.51319\,(p^{0.097} - 1.297)$$

For four-stage compression:

$$\text{H. P.} = 0.72983\,(p^{0.0725} - 1.216)$$

H. P. = Horse-power per cubic foot of free air at *sea level.*

$p$ = terminal gauge pressure in pounds per square inch + 14.75.

In general:

$$\text{H. P.} = \frac{0.2218695\text{N}}{(14.75)^{\frac{0.29}{\text{N}}}}\left[p^{\frac{0.29}{\text{N}}} - (14.75)^{\frac{0.29}{\text{N}}}\right]$$

These expressions are applicable to *sea level* conditions only.

Horse-Power and Mean Effective Pressure Developed in Compressing One Cubic Foot of Free Air From Atmospheric Pressure to Various Gauge Pressures. Initial Temperature of Air in Each Cylinder 60° F. Jacket Cooling Not Considered.

| Gauge Pressure | Atmospheres Compressed | Isothermal Compression | | Adiabatic Compression | | | | | | | | Percent. of Power Saved by Two-Stage over Single-Stage Compression |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | Single-Stage | | | | Two-Stage | | | | |
| | | M. E. P. | Horse-Power | M. E. P. Theoretical | M. E. P. 15% Friction Included | H. P. Theoretical | H. P. 15% Friction Included | M. E. P. Theoretical | M. E. P. 15% Friction Included | H. P. Theoretical | H. P. 15% Friction Included | |
| 5 | 1.34 | 4.13 | .018 | 4.46 | 5.12 | .019 | .022 | ...... | ...... | .... | ... | .... |
| 10 | 1.68 | 7.57 | .033 | 8.21 | 9.44 | .036 | .041 | ...... | ...... | .... | .... | .... |
| 15 | 2.02 | 11.02 | .048 | 11.46 | 13.17 | .050 | .057 | ...... | ...... | .... | .... | .... |
| 20 | 2.36 | 12.62 | .055 | 14.30 | 16.44 | .062 | .071 | ...... | ...... | .... | .... | .... |
| 25 | 2.70 | 14.68 | .064 | 16.94 | 19.47 | .074 | .085 | ...... | ...... | .... | .... | .... |
| 30 | 3.04 | 16.30 | .071 | 19.32 | 22.21 | .084 | .096 | ...... | ...... | .... | .... | .... |
| 35 | 3.38 | 17.90 | .078 | 21.50 | 24.72 | .094 | .108 | ...... | ...... | ... | .... | .... |
| 40 | 3.72 | 19.28 | .084 | 23.53 | 27.05 | .103 | .118 | ...... | ...... | .... | .... | .... |
| 45 | 4.06 | 20.65 | .090 | 25.40 | 29.21 | .111 | .127 | ...... | ...... | .... | .... | .... |
| 50 | 4.40 | 21.80 | .095 | 27.23 | 31.31 | .119 | .136 | ...... | ...... | .... | .... | .... |
| 55 | 4.74 | 22.95 | .100 | 28.90 | 33.23 | .126 | .145 | ...... | ...... | .... | .... | .... |
| 60 | 5.08 | 23.90 | .104 | 30.53 | 35.10 | .133 | .153 | ...... | ...... | .... | ... | .... |
| 65 | 5.42 | 24.80 | .108 | 32.10 | 36.91 | .140 | .161 | ...... | ...... | .... | .... | .... |
| 70 | 5.76 | 25.70 | .112 | 33.57 | 38.59 | .146 | .168 | 29.31 | 33.71 | .128 | .147 | 12.3 |
| 75 | 6.10 | 26.62 | .116 | 35.00 | 40.25 | .153 | .175 | 30.43 | 34.99 | .133 | .153 | 13.1 |
| 80 | 6.44 | 27.52 | .120 | 36.36 | 41.80 | .159 | .182 | 31.44 | 36.15 | .137 | .158 | 13.6 |
| 85 | 6.78 | 28.21 | .123 | 37.63 | 43.27 | .164 | .189 | 32.46 | 37.22 | .142 | .163 | 13.7 |
| 90 | 7.12 | 28.93 | .126 | 38.89 | 44.71 | .169 | .195 | 33 37 | 38.36 | .145 | .167 | 14.2 |
| 95 | 7.46 | 29.60 | .129 | 40.11 | 46.12 | .175 | .201 | 34.28 | 39.41 | .149 | .172 | 14.8 |
| 100 | 7.80 | 30.30 | .132 | 41.28 | 47.46 | .180 | .207 | 35.20 | 40.48 | .153 | .176 | 15.0 |
| 110 | 8.48 | 31.42 | .137 | 43.56 | 50.09 | .190 | .218 | 36.82 | 42.34 | .161 | .185 | 15.2 |
| 120 | 9.16 | 32.60 | .142 | 45.69 | 52.53 | .199 | .229 | 38.44 | 44.20 | .168 | .193 | 15.5 |
| 130 | 9.84 | 33.75 | .147 | 47.72 | 54.87 | .208 | .239 | 39.86 | 45.83 | .174 | .200 | 16.3 |
| 140 | 10.52 | 34.67 | .151 | 49.64 | 57.08 | .216 | .249 | 41.28 | 47.46 | .180 | .207 | 16.6 |
| 150 | 11.20 | 35.59 | .155 | 51.47 | 59.18 | .224 | .258 | 42.60 | 48.99 | .186 | .214 | 16.9 |
| 160 | 11.88 | 36.30 | .158 | 53.70 | 61.80 | .234 | .269 | 43.82 | 50.39 | .191 | .219 | 18.4 |
| 170 | 12.56 | 37.20 | .162 | 55.60 | 64.00 | .242 | .278 | 44.93 | 51.66 | .196 | .225 | 19.1 |
| 180 | 13.24 | 38.10 | .166 | 57.20 | 65.80 | .249 | .286 | 46.05 | 52.95 | .201 | .231 | 19.3 |
| 190 | 13.92 | 38.80 | .169 | 58.80 | 67.70 | .256 | .294 | 47.16 | 54.22 | .206 | .236 | 19.5 |
| 200 | 14.60 | 39.50 | .172 | 60.40 | 69.50 | .263 | .303 | 48.18 | 55.39 | .210 | .241 | 20.1 |
| 250 | 18.00 | 42.70 | .186 | 67.00 | 77.10 | .292 | .336 | 52.84 | 60.76 | .230 | .264 | 21.2 |
| 300 | 21.40 | 45.30 | .197 | 72.80 | 83.80 | .317 | .365 | 56.70 | 65.20 | .247 | .283 | 22.1 |
| 350 | 24.80 | 47.30 | .206 | 78.50 | 90.40 | .342 | .394 | 60.15 | 69.16 | .262 | .301 | 23.4 |
| 400 | 27.20 | 49.20 | .214 | 83.50 | 96.20 | .364 | .418 | 63.19 | 72.65 | .276 | .317 | 24.2 |
| 450 | 31 70 | 51.20 | .223 | 87.50 | 100.80 | .381 | .438 | 65.93 | 75.81 | .287 | .329 | 24.7 |
| 500 | 35.00 | 52.70 | .229 | 91.40 | 105.20 | .398 | .458 | 68.46 | 78.72 | .298 | .342 | 25.1 |
| 1000 | 69.04 | 62.40 | .272 | 122.90 | 141.50 | .535 | .615 | 86.80 | 100 00 | .378 | .435 | 29.4 |
| 1200 | 82.65 | 65.00 | .283 | 132.00 | 151.80 | .574 | .661 | 91.20 | 105.00 | .397 | .456 | 30.8 |
| 1400 | 96.25 | 67.00 | .292 | 140.00 | 161.00 | .610 | .702 | 95.50 | 109.90 | .416 | .478 | 31.8 |
| 1600 | 109.80 | 69.10 | .301 | 147.50 | 169.00 | .643 | .740 | 98.80 | 113.80 | .430 | .495 | 33.1 |
| 1800 | 123.40 | 71.00 | .309 | 154.00 | 177.00 | .672 | .773 | 102.20 | 117.70 | .445 | .512 | 33.9 |
| 2000 | 137.10 | 72.30 | .315 | 160.50 | 185.00 | .700 | .805 | 104.50 | 120.20 | .455 | .524 | 35.0 |

# ALTITUDE COMPRESSION.

Horse-Power Developed in Compressing Adiabatically One Cubic Foot of Free Air from Atmospheric Pressures at Altitudes to Various Gauge Pressures—Initial Temperature of Air in Each Cylinder 60° F., Jacket Cooling not Considered.

| Altitude in Feet | Atmospheric Pressure | | Horse Power Developed | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | Single-Stage Compression | | | | | | Two-Stage Compression | | | | | | | | | |
| | | | Pounds Square Inch Gauge Pressure | | | | | | Pounds Square Inch Gauge Pressure | | | | | | | | | |
| | | | 60 | | 80 | | 100 | | 60 | | 80 | | 100 | | 125 | | 150 | |
| | Inches of Mercury | Pounds per Square Inch | Theoretical | 15% Friction Included | Theoretical | 15% Friction Included | Theoretical | 15% Friction Included | Theoretical | 15% Friction Included | Theoretical | 15% Friction Included | Theoretical | 15% Friction Included | Theoretical | 15% Friction Included | Theoretical | 15% Friction Included |
| 0 | 30.05 | 14.75 | .1333 | .1533 | .1586 | .1824 | .1804 | .2075 | .1177 | .1354 | .1374 | .1580 | .1535 | .1765 | .1708 | .1964 | .1859 | .2138 |
| 1000 | 28.91 | 14.19 | .1314 | .1512 | .1562 | .1797 | .1775 | .2043 | .1158 | .1332 | .1350 | .1553 | .1507 | .1733 | .1676 | .1927 | .1823 | .2096 |
| 2000 | 27.81 | 13.65 | .1295 | .1490 | .1538 | .1769 | .1746 | .2010 | .1139 | .1310 | .1326 | .1525 | .1479 | .1701 | .1643 | .1890 | .1787 | .2054 |
| 3000 | 26.75 | 13.13 | .1276 | .1469 | .1514 | .1742 | .1717 | .1978 | .1120 | .1288 | .1302 | .1498 | .1451 | .1669 | .1611 | .1853 | .1751 | .2012 |
| 4000 | 25.73 | 12.63 | .1257 | .1447 | .1490 | .1714 | .1688 | .1945 | .1101 | .1266 | .1278 | .1470 | .1423 | .1637 | .1578 | .1816 | .1715 | .1970 |
| 5000 | 24.75 | 12.15 | .1238 | .1426 | .1466 | .1687 | .1659 | .1913 | .1082 | .1244 | .1254 | .1443 | .1395 | .1605 | .1538 | .1779 | .1679 | .1928 |
| 6000 | 23 81 | 11 69 | .1219 | .1404 | .1442 | .1659 | .1630 | .1880 | .1063 | .1222 | .1230 | .1415 | .1367 | .1573 | .1513 | .1742 | .1643 | .1886 |
| 7000 | 22.92 | 11.25 | .1200 | .1383 | .1418 | .1632 | .1601 | .1847 | .1044 | .1200 | .1206 | .1388 | .1339 | .1541 | .1480 | .1705 | .1607 | .1844 |
| 8000 | 22.06 | 10.83 | .1181 | .1361 | .1394 | .1604 | .1572 | .1815 | .1025 | .1178 | .1182 | .1360 | .1311 | .1509 | .1448 | .1668 | .1571 | .1802 |
| 9000 | 21.25 | 10.43 | .1162 | .1340 | .1370 | .1577 | .1543 | .1782 | .1006 | .1156 | .1158 | .1333 | .1283 | .1477 | .1415 | .1631 | .1535 | .1760 |
| 10000 | 20.47 | 10.05 | .1143 | .1318 | .1346 | .1549 | .1514 | .1750 | .0987 | .1134 | .1134 | .1305 | .1255 | .1445 | .1383 | .1594 | .1499 | .1718 |
| 11000 | 19 74 | 9.69 | .1124 | .1297 | .1322 | .1521 | .1485 | .1717 | .0968 | .1112 | .1110 | .1277 | .1227 | .1413 | .1350 | .1557 | .1463 | .1676 |
| 12000 | 19.05 | 9.35 | .1105 | .1275 | .1298 | .1494 | .1456 | .1685 | .0949 | .1090 | .1086 | .1250 | .1199 | .1381 | .1318 | .1520 | .1427 | .1634 |
| 13000 | 18 39 | 9.03 | .1086 | .1254 | .1274 | .1466 | .1427 | .1652 | .0930 | .1068 | .1062 | .1222 | .1171 | .1349 | .1285 | .1483 | .1391 | .1592 |
| 14000 | 17.78 | 8.73 | .1067 | .1232 | .1250 | .1439 | .1398 | .1620 | .0911 | .1046 | .1038 | .1195 | .1143 | .1317 | .1243 | .1446 | .1355 | .1551 |
| 15000 | 17.21 | 8.45 | .1048 | .1210 | .1226 | .1411 | .1369 | .1588 | .0892 | .1024 | .1014 | .1167 | .1115 | .1285 | .1221 | .1409 | .1319 | .1508 |

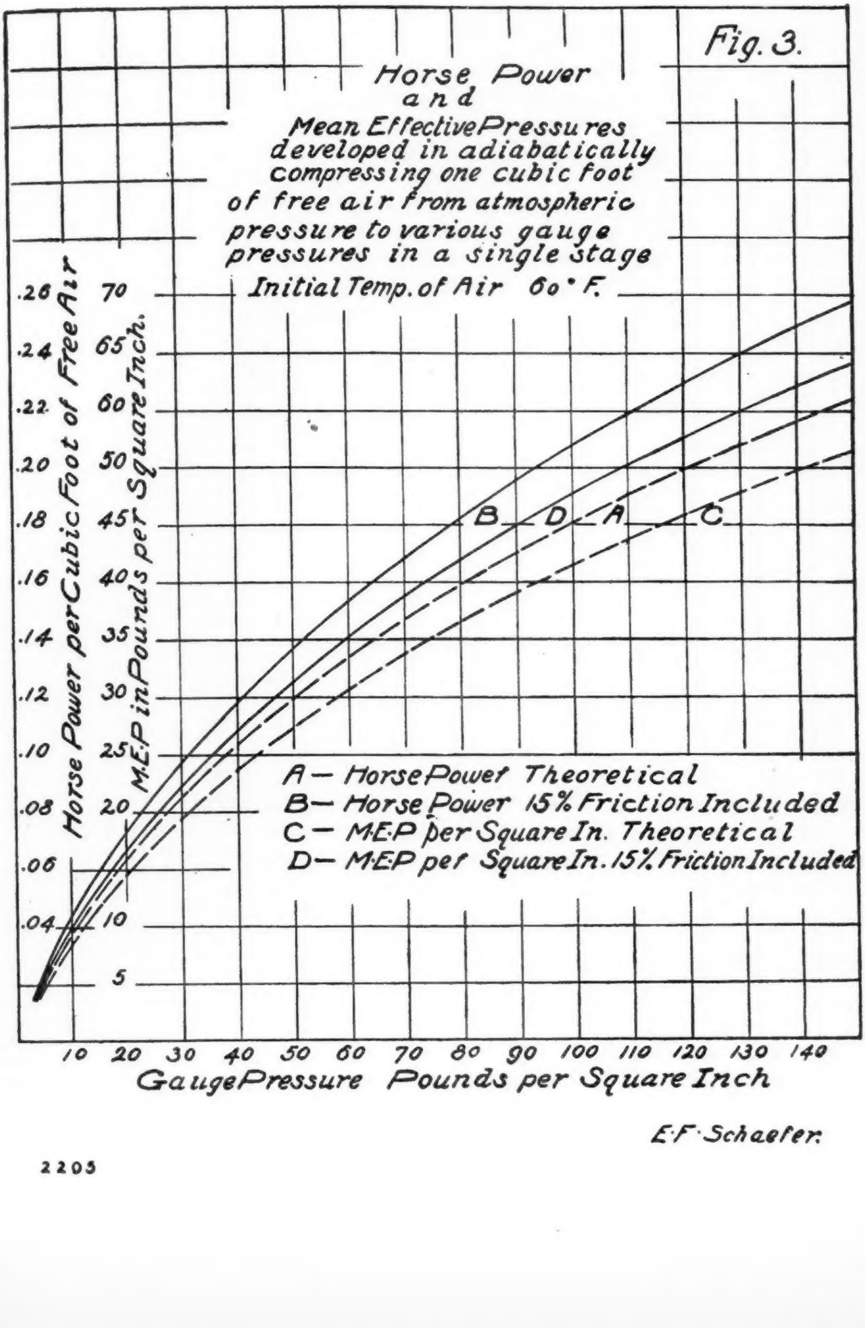
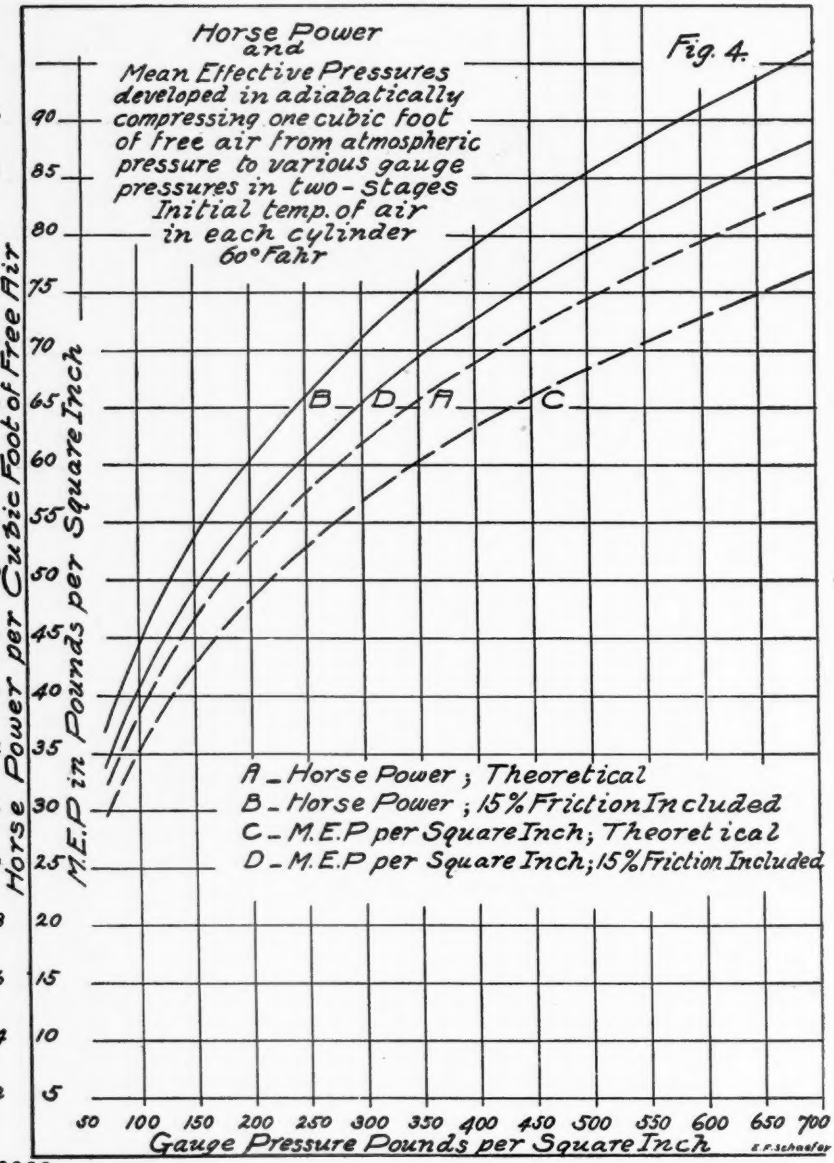


Fig. 3.
Horse Power
and
Mean Effective Pressures
developed in adiabatically
compressing one cubic foot
of free air from atmospheric
pressure to various gauge
pressures in a single stage
Initial Temp. of Air 60° F.
Horse Power per Cubic Foot of Free Air
M.E.P in Pounds per Square Inch.
.26 .24 .22 .20 .18 .16 .14 .12 .10 .08 .06 .04
70 65 60 50 45 40 35 30 25 20 10 5
B D A C
A — Horse Power Theoretical
B — Horse Power 15% Friction Included
C — M.E.P per Square In. Theoretical
D — M.E.P per Square In. 15% Friction Included
10 20 30 40 50 60 70 80 90 100 110 120 130 140
Gauge Pressure Pounds per Square Inch
E.F. Schaefer.


Horse Power and Mean Effective Pressures developed in adiabatically compressing one cubic foot of free air from atmospheric pressure to various gauge pressures in two-stages Initial temp. of air in each cylinder 60° Fahr
Fig. 4.
B D A C
A - Horse Power ; Theoretical
B - Horse Power ; 15% Friction Included
C - M.E.P per Square Inch ; Theoretical
D - M.E.P per Square Inch ; 15% Friction Included
Horse Power per Cubic Foot of Free Air
M.E.P in Pounds per Square Inch
.38 .36 .34 .32 .30 .28 .26 .24 .22 .20 .18 .16 .14 .12 .10 .08 .06 .04 .02
90 85 80 75 70 65 60 55 50 45 40 35 30 25 20 15 10 5
50 100 150 200 250 300 350 400 450 500 550 600 650 700
Gauge Pressure Pounds per Square Inch
E. F. Schaefer

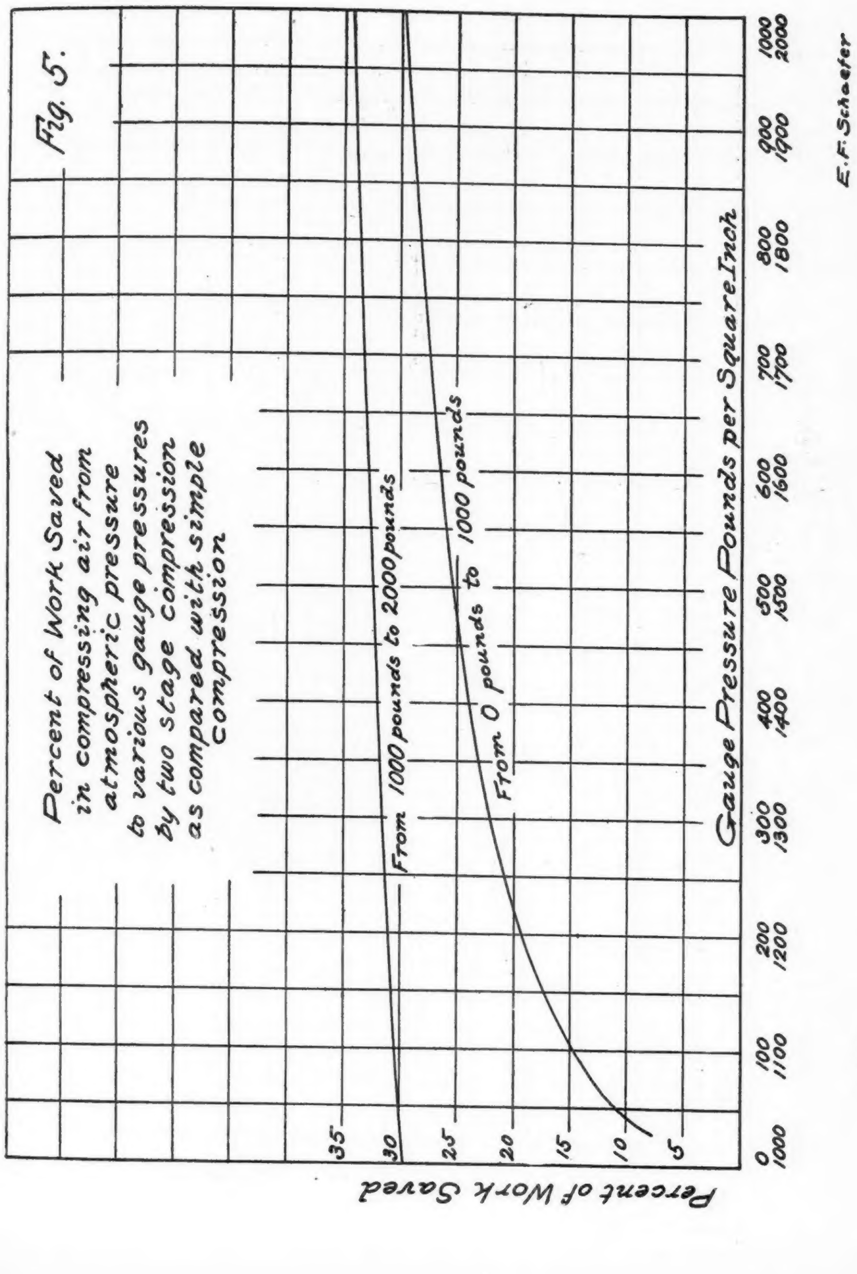
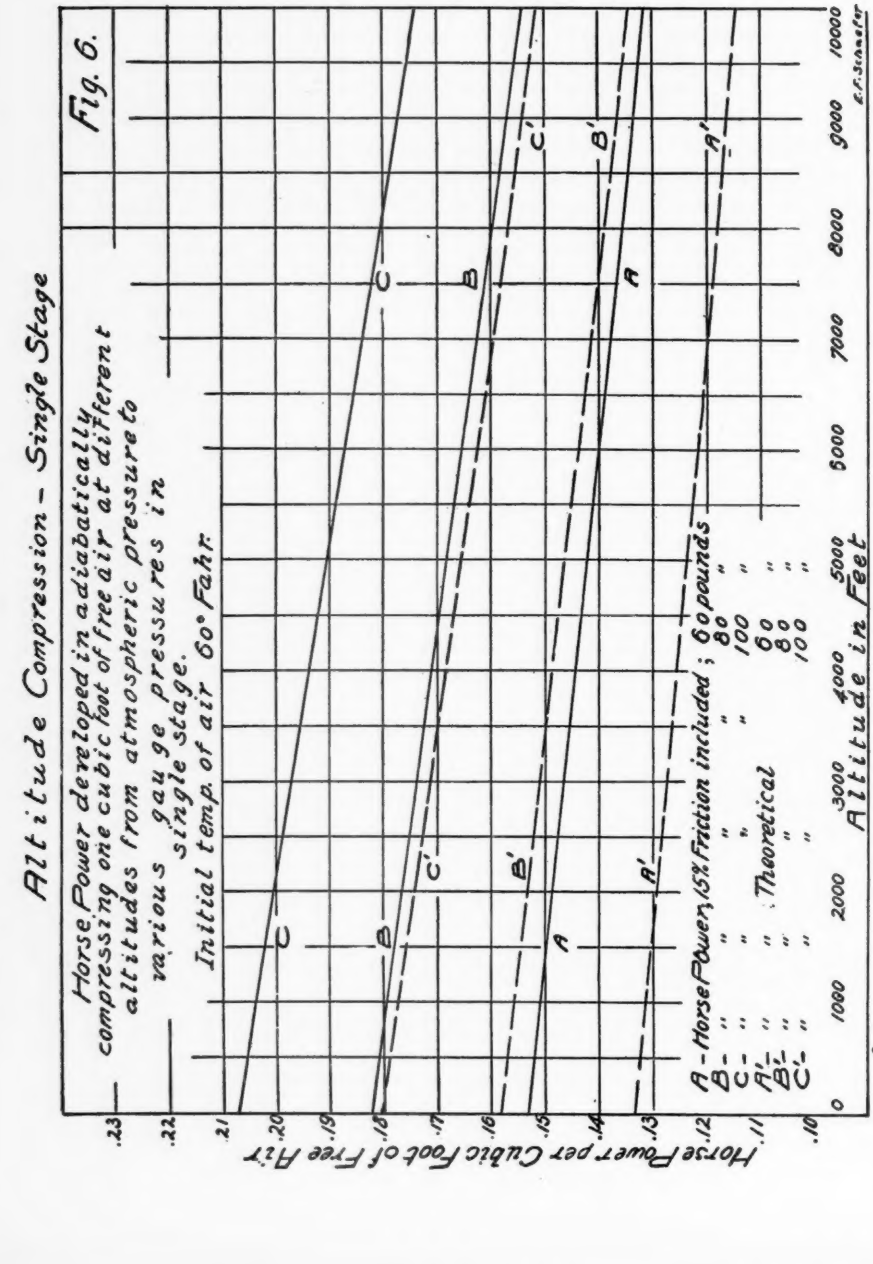
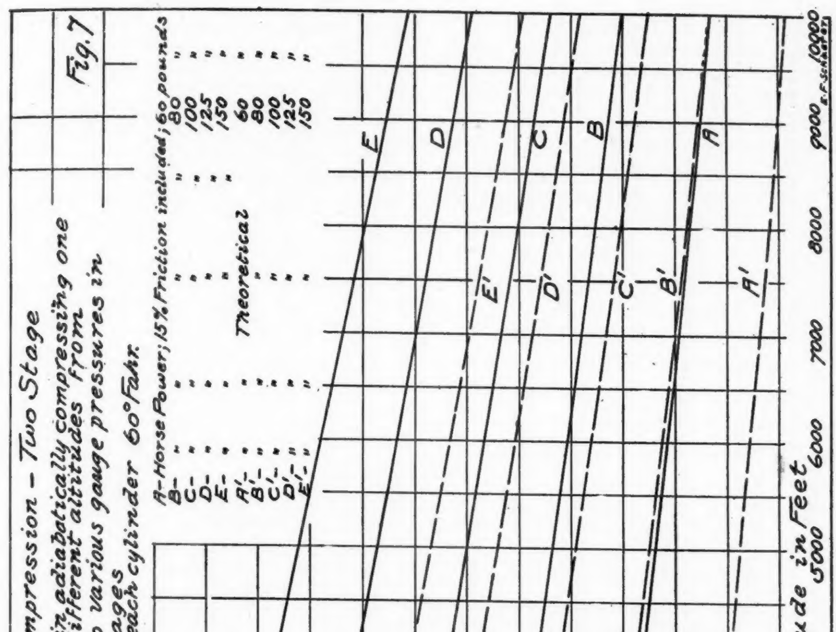


Fig. 5.
Percent of Work Saved in compressing air from atmospheric pressure to various gauge pressures by two stage compression as compared with simple compression
From 1000 pounds to 2000 pounds
From 0 pounds to 1000 pounds
0 5 10 15 20 25 30 35
Percent of Work Saved
0 1000
100 1100
200 1200
300 1300
400 1400
500 1500
600 1600
700 1700
800 1800
900 1900
1000 2000
Gauge Pressure Pounds per Square Inch
E. F. Schaefer


Altitude Compression - Single Stage
Fig. 6.
Horse Power developed in adiabatically compressing one cubic foot of free air at different altitudes from atmospheric pressure to various gauge pressures in single stage.
Initial temp. of air 60° Fahr.
A - Horse Power, 15% Friction included; 60 pounds
B - " " " " ; 80 "
C - " " " " ; 100 "
A' - " " Theoretical 60 "
B' - " " " 80 "
C' - " " " 100 "
A
B
C
A'
B'
C'
Horse Power per Cubic Foot of Free Air
.23
.22
.21
.20
.19
.18
.17
.16
.15
.14
.13
.12
.11
.10
Altitude in Feet
0
1000
2000
3000
4000
5000
6000
7000
8000
9000
10000
2208


Altitude Compression - Two Stage
Horse Power developed in adiabatically compressing one cubic foot of free air at different altitudes from atmospheric pressure to various gauge pressures in two stages
Initial temp. of air in each cylinder 60° Fahr.
Fig. 7
A - Horse Power; 15% Friction included; 60 pounds
B - " " " " 80 "
C - " " " " 100 "
D - " " " " 125 "
E - " " " " 150 "
A' - " " Theoretical 60 "
B' - " " " 80 "
C' - " " " 100 "
D' - " " " 125 "
E' - " " " 150 "
Altitude in Feet
0
1000
2000
3000
4000
5000
6000
7000
8000
9000
10000
Horse Power per Cubic Foot of Free Air
.24
.23
.22
.21
.20
.19
.18
.17
.16
.15
.14
.13
.12
.11
.10
E
D
C
B
A
E'
D'
C'
B'
A'
E. F. Schaefer

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If the horse-powers for altitude conditions are required substitute in the above general formulae the proper altitude atmospheric pressure in place of 14.75.

Below are found formulae giving directly the horse-powers, for *altitude* conditions, for both theoretical and practical conditions. The only datum needed is the elevation in feet.

Let $E$ = elevation in feet

$HP_T$ = theoretical horse-power

$HP_F$ = horse-power including 15 per cent. friction

SINGLE STAGE.

For 60-pound Gauge Pressure.

$$HP_T = 0.1333 - 0.0000019E$$
$$HP_F = 0.1533 - 0.00000215E$$

For 80-pound Gauge Pressure.

$$HP_T = 0.1586 - 0.0000024E$$
$$HP_F = 0.1824 - 0.00000275E$$

For 100-pound Gauge Pressure.

$$HP_T = 0.1804 - 0.0000029E$$
$$HP_F = 0.2075 - 0.00000325E$$

TWO STAGES.

For 60-pound Gauge Pressure.

$$HP_T = 0.1177 - 0.0000019E$$
$$HP_F = 0.1354 - 0.0000022E$$

For 80-pound Gauge Pressure.

$$HP_T = 0.1374 - 0.0000024E$$
$$HP_F = 0.1580 - 0.00000275E$$

For 100-pound Gauge Pressure.

$$HP_T = 0.1535 - 0.0000028D$$
$$HP_F = 0.1765 - 0.0000032E$$

For 125-pound Gauge Pressure.

$$HP_T = 0.1708 - 0.00000325E$$
$$HP_F = 0.1964 - 0.0000037E$$

For 150-pound Gauge Pressure.

$$HP_T = 0.1859 - 0.0000036E$$
$$HP_F = 0.2138 - 0.0000042E$$

The preceding formulae were employed in securing the data, represented in graphical form, in Fig. 3 to Fig. 7, inclusive.

Fig. 3 gives the mean effective pressures and horse-powers for single-stage operations in adiabatical compressing and in delivering one cubic foot of free air from atmospheric pressure to various gauge pressures. In these calculations it was assumed that the temperature of the intake air was 60 degrees Fahrenheit, which is the normal condition. Curve A given the theoretical horse-power necessary without considering fractional losses. In curve B, however, we find an allowance made for friction. The extra horse-power necessary when considering the work consumed by the movement of the piston, crosshead and crank-pins, etc., is less than 15 per cent., but we allow 15 per cent. to be on the safe side. Curves C and D represent the M. E. P. for theoretical and for frictional conditions, respectively.

Fig. 4 represents also the horse-powers and M. E. P.s for both theoretical and actual conditions. It is, however, devoted to two-stage compression and the initial temperature of the air, as it enters both the low pressure and high pressure cylinders, is taken as 60 degrees Fahrenheit.

Fig. 5 shows the per cent. of work saved in compressing air from atmospheric pressure to various gauge pressures by two-stage over simple compression. The values are found as follows:

Let $a$ = work done in compressing in one stage to pressure $p$.

Let $b$ = work done in compressing in two stages to pressure $p$.

Per cent. of power saved = $(a-b) \div a$.

Suppose, for example, that the per cent. of power saving is wanted when compressing to 100 pounds gauge. For 100 pounds we get from Fig. 3 0.207 horse-power, and from Fig. 4 we get 0.176 horsepower. Then $(0.207 - 0.176) \div 0.207 = 15$ per cent.

Fig. 6 gives the horse-powers necessary to compress one cubic foot of free air adiabatically at different *altitudes* from atmospheric pressure to various gauge pressures (60 pounds, 80 pounds and 100 pounds) in simple compression. Curves A, B and C include frictional losses and $A^1$, $B^1$ and $C^1$ are theoretical simply.

Fig. 7 is identical with Fig. 6, except that it is devoted to compound compression. The various terminal pressures are 60 pounds, 80 pounds, 100 pounds, 125 pounds and 150 pounds per square inch gauge.

The usefulness of the above formulae must now be apparent.

In addition to the foregoing solutions of the formulae expressing adiabatic conditions, it might also be of interest, as a basis of comparison, to know what forms the equations assume when compression

is purely isothermal. The isothermal compression or expansion is the ideal. It is, of course, impossible, but the actual state is somewhere between the isothermal and adiabatic and usually close to the latter.

As we have to deal usually with work done in compression and transference, minus the work done by the atmosphere, the area or work is represented by the indicator card Fig. 8 and is expressed by:

$$\text{Work done} = CA^1FHC = ECA^1D^1 - ECB^1D^1 + (A^1FGD^1 - B^1HGD^1) \quad \text{.........(16)}$$

For isothermal compression:

$$pv = p_1 v_1 = p_2 v_2 = \text{\&c.} \quad \text{..........(17)}$$

$$p = \frac{p_1 v_1}{v}$$

Then:

$$ECA^1D^1 = p_1 v_1 \log_e\left(\frac{p_2}{p_1}\right) \quad \text{.......(20)}$$

$$= p_1 v_1 \times 2.3 \log\frac{p_2}{p_1} \quad \text{.....(21)}$$

Now:

$$\left.\begin{array}{l} p_1 v_1 = CHGE \\ p_2 v_2 = A^1FGD^1 \end{array}\right\} \quad \text{................ (22}$$

and since $p_1 v_1 = p_2 v_2$ we have:

$$CHGE = A^1FGD^1$$

or

$$CHGE - B^1HGD^1 = A^1FGD^1 - B^1HGD^1$$

$$ECB^1D^1 = A^1FHB^1 \quad \text{.....(23)}$$

Substituting in (16)

$$\text{Work done} = CA^1FHC = ECA^1D^1 - ECB^1D^1 + A^1FHB^1 \quad \text{....(24)}$$

$$\text{Work done} = CA^1FHC = ECA^1D^1 \quad \text{....(25)}$$

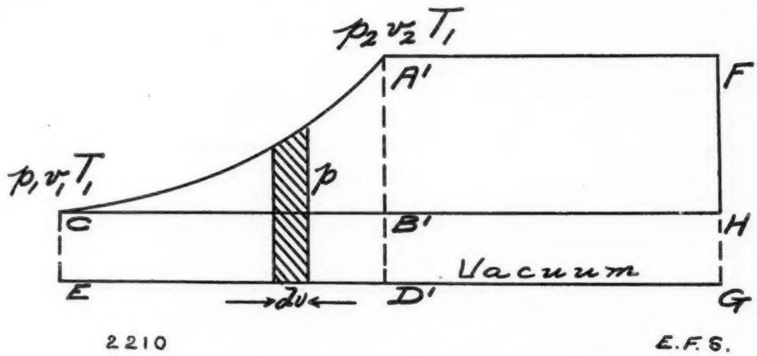


Fig. 8

From Calculus:

$$ECA^1D^1 = \int p\,dv \quad \text{......(18)}$$

$$= \int \frac{p_1 v_1}{v}\,dv$$

$$= p_1 v_1 \int_{v_2}^{v_1} \frac{dv}{v}$$

$$= p_1 v_1 \log_e\left(\frac{v_1}{v_2}\right) \quad \text{........(19)}$$

In which $e$ is the base of the Naperian system of logarithms.

From (17)

$$\frac{v_1}{v_2} = \frac{p_2}{p_1}$$

and equating with (21)

$$\text{Work done} = 2.3\, p_1 v_1 \log\left(\frac{p_2}{p_1}\right) \quad \text{.......(26)}$$

Therefore, in isothermal compression the net work of compression and transference is equal to the total work of compression.

In isothermal compression the horsepower necessary per cubic foot of free air will be the same if we employ one, two, three or more stages in which to do the work.

Divide (26) by $v_1$ and the expression will represent the M. E. P. per stroke or

$$\text{M. E. P.} = 2.3\, p_1 \log \left(\frac{p_2}{p_1}\right) \ldots\ldots (27)$$

$$\text{H. P.} = \frac{144}{33000} \text{M. E. P.}$$

$$= 0.00436 \text{ M. E. P.} \ldots\ldots (28)$$

in which the pressures are reduced to pounds per square inch.

It is now my purpose to show that when air cylinders are proportioned according to the law

$$r_1 = r_2 = r_3 = \text{etc.} = r_n = \sqrt[n]{R}$$

the best conditions are possible and maximum saving in horse-power is the result. (Fig. 9.)

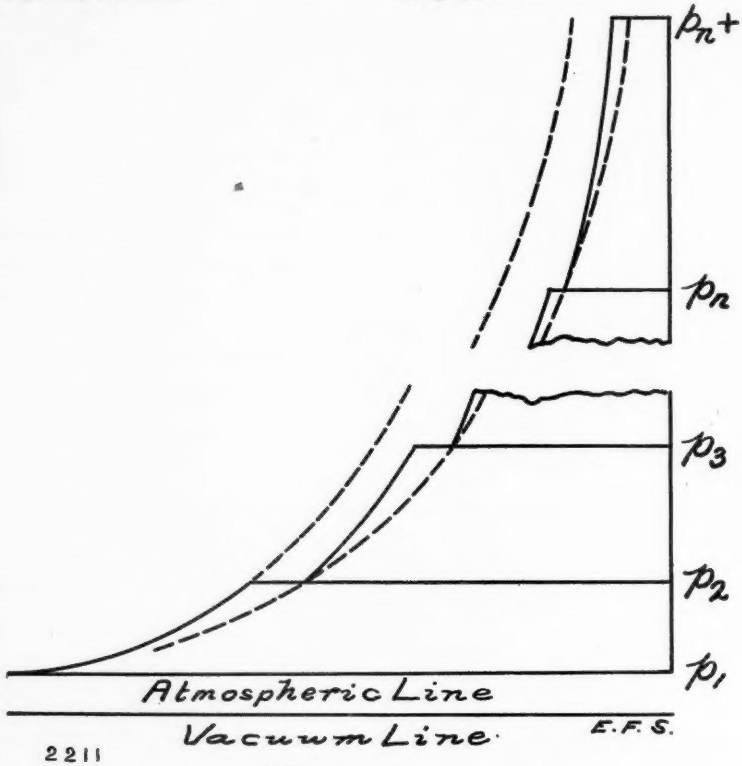


Fig. 9

*Example III.*

Required the horse-power necessary to compress isothermally one cubic foot of free air to 90 pounds per square inch gauge pressure.

$p_1$ = 14.75 pounds per square inch.
$p_2$ = 14.75 + 90 pounds per square inch.

$$\text{M. E. P.} = 2.3 \times 14.75 \times \log\left(\frac{104.75}{14.75}\right)$$

$$= 28.88$$

H. P. = 0.00436 M. E. P. = 0.1259 per cubic foot of free air.

Letting:

$$\frac{p_n + 1}{p_1} = R$$

$$\frac{p_2}{p_1} = r_1$$

$$\frac{p_3}{p_2} = r_2$$

$$\frac{p_n + 1}{p_n} = r_n$$

Assuming:

$$r_1 = r_2 = \text{etc.} = r_n = \sqrt[n]{R}$$

We have:

$$\frac{p_2}{p_1}\cdot\frac{p_3}{p_2}\cdots\cdots\cdots\frac{p_n+1}{p_n}=\frac{p_n+1}{p_1}=R$$

$$r_1\cdot r_2\cdots\cdots\cdots r_n=R$$

$$r_n=\frac{R}{r_1^{\,n-1}}$$

$$r_3=\frac{R}{r_1^{\,n-1}}$$

$$r_2=\frac{R}{r_1^{\,n-1}}$$

From equation (12)

$$W_{total}=3.45\,p_1\,v_1\,(r_1^{0.29}-1)+3.45\,p_2\,v_2\,(r_2^{0.29}-1)+\text{etc}+3.45\,p_n\,v_n\,(r_n^{0.29}-1)$$

$$\text{But } p_1\,v_1=p_2\,v_2=\text{etc}=p_n\,v_n$$

$$W_{total}=3.45\,p_1\,v_1\,[(r_1^{0.29}-1)+(r_2^{0.29}-1)+\text{etc}+(r_n^{0.29}-1)]$$

$$=3.45\,p_1\,v_1[r_1^{0.29}+r_2^{0.29}+\text{etc}+r_n^{0.29}-n]$$

$$=3.45\,p_1\,v_1[(n-1)\,r_1^{0.29}+R^{0.29}\,r_1^{-(n-1)\,0.29}-n]$$

$$=3.45\,p_1\,v_1[(n-1)\,r_1^{0.29}+R^{0.29}\,r_1^{0.29-0.29\,n}-n]$$

$$\frac{d\,W}{d\,r_1}=(n-1)\,0.29\,r_1^{-0.71}+(0.29-0.29\,n)\,R^{0.29}\,r_1^{-0.71-0.29\,n}=0$$

$$=(0.29\,n-0.29)\,r_1^{-0.71}-(0.29\,n-0.29)\,R^{0.29}\,r_1^{-0.71-0.29\,n}=0$$

Divide by $(0.29\,n-0.29)\,r_1^{-0.71}$

$$1-R^{0.29}\,r_1^{-0.29\,n}=0$$

$$R^{0.29}\,r_1^{-0.29\,n}=1$$

$$r_1^{-0.29\,n}=\frac{1}{R^{0.29}}$$

$$r_1^{-0.29\,n}=R^{-0.29}$$

$$r_1=R^{\frac{1}{n}}$$

$$r_1=\sqrt[n]{R}$$

$$\frac{d^2\,W_t}{d\,r_1^2}=-0.71\,r_1^{-1.71}-(-0.71-0.29\,n)\,R^{0.29}\,r_1^{-1.71-0.29\,n}$$

$$=-0.71\,r_1^{-1.71}+(0.71+0.29\,n)\,r_1^{0.29\,n}\,r_1^{-1.71-0.29\,n}$$

$$=-0.71\,r_1^{-1.71}+(0.71+0.29n)\,r_1^{-1.71}$$

$$=-\frac{0.71}{r_1^{1.71}}+\frac{0.71+0.29\,n}{r_1^{1.71}}$$

It is evident that the expression will always be positive. The differential is of an even order and positive showing.

$$r_1=\sqrt[n]{R}$$

corresponds to a minimum.

In other words, when:

$$r_1=r_2=r_3=\text{etc}=r_n=\sqrt[n]{R}$$

the work is equally divided between the cylinders and the maximum saving in power is effected.

---

### Recapitulation.

**$p_i$ = absolute intake pressure in pounds per square inch.**

**$p_t$ = absolute discharge pressure in pounds per square inch.**

**N = number of stages in which compression is to be accomplished.**

**M. E. P.$_T$ = theoretical mean effective pressure.**

**H. P.$_T$ = theoretical horse-power.**

**H. P.$_F$ = horse-power including 15 per cent. friction.**

**E = altitude in feet.**

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*For Isothermal or Adiabatic Compression.*

**Horse-Power = 0.00436 M. E. P.$_T$ .....(A)**

---

*Isothermal Compression for any number of stages.*

**M. E. P.$_T$ = $2.3\,p_i\log\left(\frac{p_t}{p_i}\right)$..........(B)**

---

*Adiabatic Compression.*

For ONE stage

**M. E. P.$_T$ = $3.45\,p_i\left[\left(\frac{p_t}{p_i}\right)^{0.29}-1\right]$.... (C)**

For TWO stages

**M. E. P.$_T$ = $6.90\,p_i\left[\left(\frac{p_t}{p_i}\right)^{0.145}-1\right]$....(D)**

For THREE stages

**M. E. P.$_T$ = $10.35\,p_i\left[\left(\frac{p_t}{p_i}\right)^{0.097}-1\right]$... (E)**

For FOUR stages

**M. E. P.$_T$ = $13.80\,p_i\left[\left(\frac{p_t}{p_i}\right)^{0.0725}-1\right]$... (F)**

For FIVE stages

**M. E. P.$_T$ = $17.25\,p_i\left[\left(\frac{p_t}{p_i}\right)^{0.058}-1\right]$....(G)**

In general:

$$\text{M. E. P.}_T = 3.45\, N p_1 \left[ \left(\frac{p_t}{p_i}\right)^{\frac{0.29}{N}} - 1 \right] \quad \text{(H)}$$

*Adiabatic Compression for Sea Level Conditions.*

For ONE stage

$$\text{H. P.}_T = 0.10163\,(p_t^{\,0.29} - 2.183) \quad \text{(I)}$$

For TWO stages

$$\text{H. P.}_T = 0.30043\,(p_t^{\,0.145} - 1.477) \quad \text{(J)}$$

For THREE stages

$$\text{H. P.}_T = 0.51319\,(p_t^{\,0.097} - 1.297) \quad \text{(K)}$$

For FOUR stages

$$\text{H. P.}_T = 0.72983\,(p_t^{\,0.0725} - 1.216) \quad \text{(L)}$$

In general:

$$\text{H. P.}_T = \frac{0.2218695\,N}{(14.75)^{\frac{0.29}{N}}} \left[ p_t^{\frac{0.29}{N}} - (14.75)^{\frac{0.29}{N}} \right] \quad \text{(M)}$$

*Adiabatic Compression for Altitude Conditions.*

SINGLE STAGE.

For 60-pound Gauge Pressure.

$$\text{H. P.}_T = 0.1333 - 0.0000019E \quad \text{(N)}$$
$$\text{H. P.}_F = 0.1533 - 0.00000215E \quad \text{(O)}$$

For 80-pound Gauge Pressure.

$$\text{H. P.}_T = 0.1586 - 0.0000024E \quad \text{(P)}$$
$$\text{H. P.}_F = 0.1824 - 0.00000275E \quad \text{(Q)}$$

For 100-pound Gauge Pressure.

$$\text{H. P.}_T = 0.1804 - 0.0000029E \quad \text{(R)}$$
$$\text{H. P.}_F = 0.2075 - 0.00000325E \quad \text{(S)}$$

TWO STAGES.

For 60-pound Gauge Pressure.

$$\text{H. P.}_T = 0.1177 - 0.0000038E \quad \text{(T)}$$
$$\text{H. P.}_F = 0.1354 - 0.0000022E \quad \text{(U)}$$

For 80-pound Gauge Pressure.

$$\text{H. P.}_T = 0.1374 - 0.0000024E \quad \text{(V)}$$
$$\text{H. P.}_F = 0.1580 - 0.00000275E \quad \text{(W)}$$

For 100-pound Gauge Pressure.

$$\text{H. P.}_T = 0.1535 - 0.0000028E \quad \text{(X)}$$
$$\text{H. P.}_F = 0.1765 - 0.0000032E \quad \text{(Y)}$$

For 125-pound Gauge Pressure.

$$\text{H. P.}_T = 0.1708 - 0.00000325E \quad \text{(AA)}$$
$$\text{H. P.}_F = 0.1964 - 0.0000037E \quad \text{(BB)}$$

For 150-pound Gauge Pressure.

$$\text{H. P.}_T = 0.1859 - 0.0000036E \quad \text{(CC)}$$
$$\text{H. P.}_F = 0.2138 - 0.0000042F \quad \text{(DD)}$$

*Stage Compression.*

Condition for best proportion

$$r_1 = r_2 = r_3 = r_N = \sqrt[N]{R} \quad \text{(EE)}$$

## Compound Air Compression.

BY L. I. WIGHTMAN, E. E.

It is well known that the heating of air produces an increase in its volume, whatever the source of the heat. The heat produced in a cylinder by compression acts to expand the air in that cylinder, whatever the speed or rate of compression. In effect this is equivalent to an increase in the volume of air to be compressed and delivered. This in turn calls for an increase in the power, to compress this apparently added volume of air; or, to put it differently, the heat of compression, in increasing the volume of air, makes it necessary to carry the air to a higher mean effective pressure in the cylinder, in order to secure finally the required volume of air at the required pressure after its temperature has fallen to that of the surrounding atmosphere, requiring, therefore, an excess of power. A consideration of these facts suggests that if some means be provided for removing the heat of compression as fast as it is produced, there will be an important reduction in the power required to raise a given weight or volume of air to a given pressure.

When air is compressed in a cylinder without the removal or escape of any of the heat produced, the compression is known as "adiabatic." When compression is carried on in such a way that heat is removed as fast as produced, the compression is "isothermal." In the first case the air delivered under pressure will be at the high terminal temperature corresponding to that pressure. In the second, the compressed air will have the same temperature at which it entered the cylinder. Adiabatic compression is the kind which all pneumatic engineers seek to avoid, while isothermal compression is the impossible ideal. The actual results secured in the best compressors are intermediate between these, but nearer to the adiabatic.

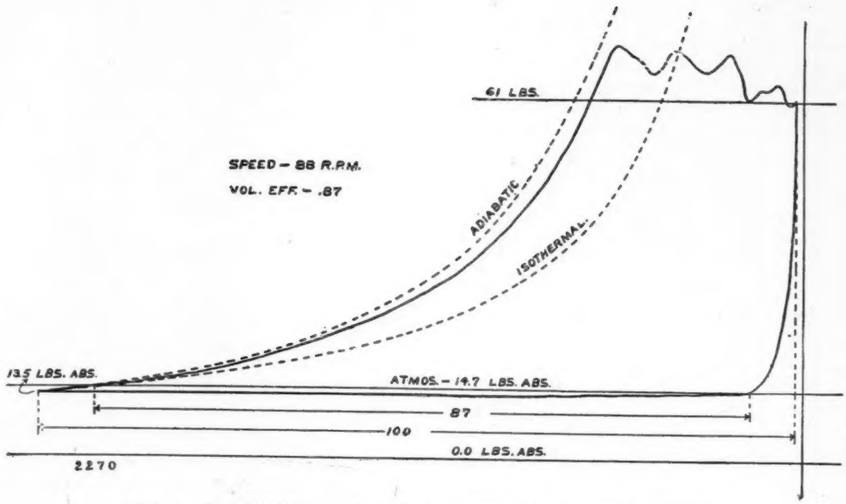



FIG. I. INDICATOR CARD FROM SINGLE-STAGE COMPRESSOR.

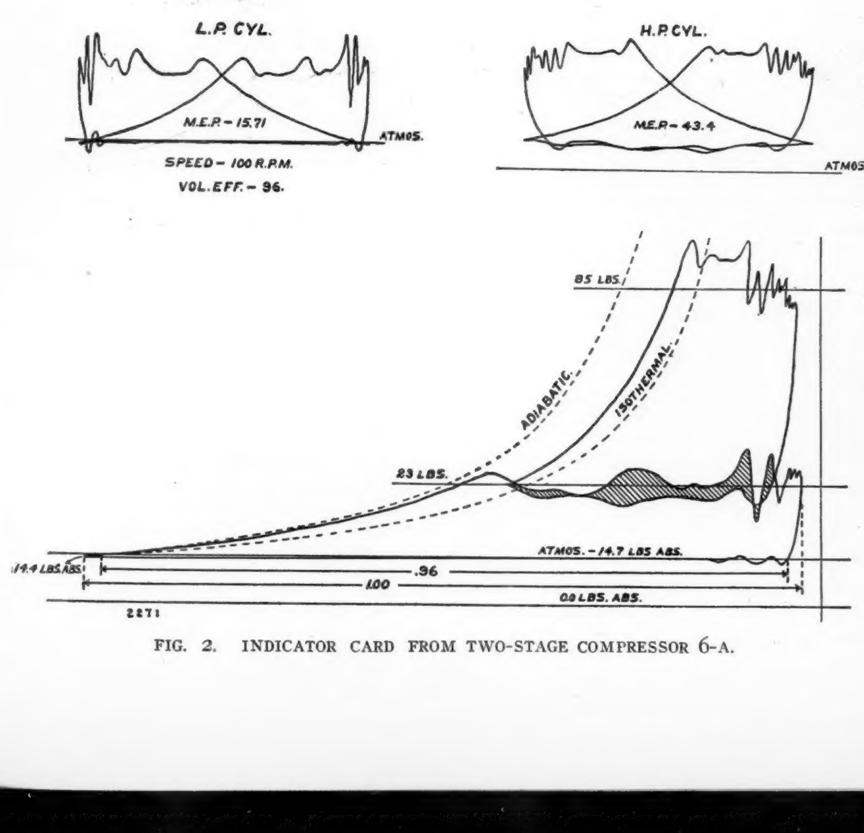



FIG. 2. INDICATOR CARD FROM TWO-STAGE COMPRESSOR 6-A.

Other things being equal, the economy of an air compressor depends on the proportion of the heat of compression which is removed as developed. Compressor efficiency, therefore, may be said to depend upon the effectiveness of the cooling devices adopted, provided what is gained here is not elsewhere wasted in whole or part. After long experience, bitter alike to makers and users, modern practice in compressor design recognizes only two practical methods of removing the heat of compression; jacket cooling and intercooling. These will be considered in order.

*Jacket Cooling.*—A brief consideration of the conditions will show that jacketed barrel cooling alone can be only a partial and very unsatisfactory solution of the problem of removing the heat developed by compression. With the piston at the beginning of its stroke, the maximum cold cylinder surface is exposed and the cylinder is filled with air at its lowest pressure and temperature. As the piston advances, pressure and temperature increase, while the effective area of cooling surface diminishes; and when the maximum pressure and temperature are attained near the end of the stroke, practically none of the cylinder wall is exposed except on the intake side of the piston; if the head is jacketed, it alone exerts any cooling influence. Furthermore, throughout the stroke only the outside layer of the air can be in contact with the cold surface and, air being a poor conductor of heat, none of the heat from the interior of the air volume is dissipated in the cooling water.

Cylinder jacketing is advisable and even essential for keeping the metal of the working parts at a low temperature, preventing the caking of lubricant upon the cylinder walls, and other evils of a hot machine. But it cannot of itself be considered as an adequate solution of the problem of cooling during compression. However, in constructions involving the use of a piston inlet tube and valve, not only the barrels, but the heads and discharge valves are chillled; and the piston and tube themselves are kept relatively cold. The air enters through a cold passage, is in contact on all sides with cold metal, and the maximum effects obtainable from jacketing alone are secured.

*Intercooling.*—If, at each of several points in the stroke the piston should be stopped for a moment and the air, already partially compressed and heated, be withdrawn long enough to be cooled by some external means to its initial temperature and then returned to the cylinder to be further compressed, it is evident that a fairly uniform temperature could be maintained in the air volume throughout the range of pressures from initial to terminal. The result would be in effect nearly that of isothermal compression. Evidently mechanical considerations forbid in practice such repeated starting and stopping of the piston; but the same results may be secured by carrying on the process of compression in several cylinders, in the first of which a certain low pressure is reached and the air at this pressure discharged through a cooling device to a second cylinder; there it is compressed to a still higher pressure and is discharged through another cooler to a third cylinder for further compression; and so on, until the required terminal pressure is secured. Such a process, developed to a practical working basis, is the method of compression in multi-stage cylinders which has become practically standard in air-compressor work for the higher pressures.

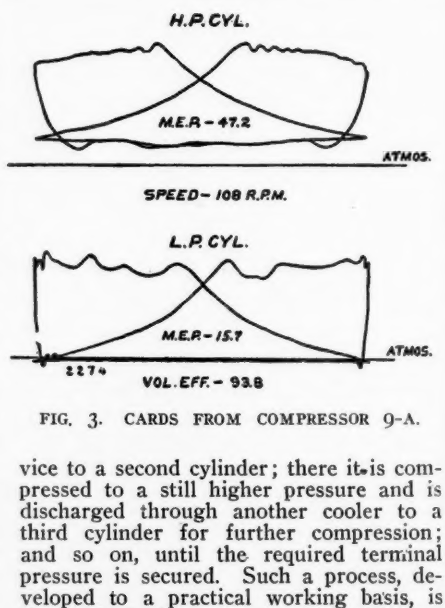


FIG. 3. CARDS FROM COMPRESSOR 9-A.

*Multi-Stage Compression.*—Theoretically, there is a gain in compound compression, whatever the pressure. But with low pressures the saving is so small as to be offset by the greater expense and complication involved in several cylinders and the losses unavoidable in the operation of

added parts. After extended experience, makers of air compressors have fixed upon 70 pounds gauge pressure as the maximum terminal pressure which can be most economically obtained in a single cylinder; and for pressures from 75 pounds up, they have adopted compound compression in two, three and four stages, the number of stages increasing with the pressure. At high altitudes, however, with large volumes and expensive fuel, this dividing line may come at a lower pressure. It is elastic and depends somewhat on the conditions.

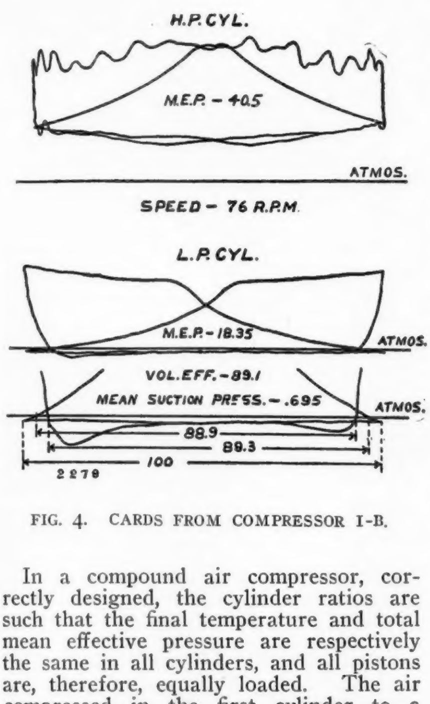



FIG. 4. CARDS FROM COMPRESSOR I-B.

In a compound air compressor, correctly designed, the cylinder ratios are such that the final temperature and total mean effective pressure are respectively the same in all cylinders, and all pistons are, therefore, equally loaded. The air compressed in the first cylinder to a pressure determined by the cylinder ratios is discharged through the outlet valves to an intercooler, where it is split up into thin streams passing over cold surfaces. The best practice provides a nest of tubes through which cold water circulates, and over and between which the stream of air passes, complete breaking-up and subdivision of the stream being secured by baffle plates and the tubes themselves. A properly designed intercooler having sufficient cooling area for the volume of air may reduce the temperature of the air compressed in the first cylinder to at least outgoing water temperature. From the intercooler this air, entering the second cylinder cold, is compressed to a higher pressure and again reaches a temperature nearly the same as that attained in the first cylinder. In two-stage machines this air will be discharged directly to the receiver without further cooling unless conditions are such as to render advisable the use of an aftercooler. In three-stage machines the second cylinder is known as the intermediate, from which the air passes to the second intercooler to undergo a second reduction of temperature, and thence enter the third cylinder for final compression to required pressure.

It is evident that multi-stage compression is in effect identical with the theoretical process suggested above, in which the compressing piston was stopped and the air cooled at intervals during the stroke. The maximum cooling effect, and therefore saving, is secured by making the intercoolers of ample proportions and providing for the splitting up of the air stream into thin sheets exposed to cooling action.

The discussion thus far has dealt with the theory of compound air compression, the conditions encountered, and the means adopted in the best practice for meeting these conditions. General statements of the gains secured by compounding have been made. It remains to discuss in detail some of the more important and specific advantages arising from stage compression.

*Reduced Power.*—The accompanying table (1) gives the theoretical percentage of work lost in the heat of compression in one, two and four stages, at various pressures. In these figures no account is taken of jacket cooling, for the reasons already stated; nor is any allowance made for certain inevitable mechanical losses.

Taking a specific example, the saving by compounding strikingly appears. Assume that compressed air is to be delivered at a pressure of 100 pounds, and a rate equivalent to 100 final effective horsepower. Referring to the table, in the second column the theoretical percentage of lost work in one-stage compression is given at 38 per cent.; but, because there is bound to be some radiation of heat,

## LOSS OF WORK DUE TO HEAT IN COMPRESSING AIR FROM ATMOSPHERIC PRESSURE TO VARIOUS GAUGE PRESSURES BY SIMPLE AND COMPOUND COMPRESSION.

Air in Each Cylinder ; Initial Temperature 60° F.

| Gauge Pressure | One-Stage | | Two-Stage | | Three-Stage | | Four-Stage | |
|---|---|---|---|---|---|---|---|---|
| | Percentage of Work Lost in Terms of | | | | | | | |
| | Isothermal Compression | Adiabatic Compression | Isothermal Compression | Adiabatic Compression | Isothermal Compression | Adiabatic Compression | Isothermal Compression | Adiabatic Compression |
| 60 | 29.9 | 23.0 | 13.4 | 11.8 | 8.6 | 7.9 | 4.7 | 4.5 |
| 70 | 30.6 | 23.4 | 14.1 | 12.4 | 8.7 | 8.0 | 6.1 | 5.7 |
| 80 | 32.7 | 24.6 | 14.7 | 12.8 | 9.7 | 8.9 | 6.4 | 6.0 |
| 90 | 34.7 | 25.8 | 16.1 | 13.8 | 10.5 | 9.5 | 7.3 | 6.8 |
| 100 | 36.7 | 26.8 | 16.9 | 14.5 | 10.9 | 9.8 | 7.8 | 7.3 |
| 125 | 41.1 | 29.2 | 18.5 | 15.6 | 11.6 | 10.4 | 8.8 | 8.1 |
| 150 | 44.8 | 30.9 | 20.1 | 16.7 | 12.3 | 10.9 | 9.1 | 8.4 |
| 200 | 51.2 | 33.9 | 22.2 | 18.1 | 14.0 | 12.3 | 10.5 | 9.5 |
| 300 | 61.2 | 37.9 | 25.7 | 20.5 | 16.6 | 14.2 | 12.0 | 10.7 |
| 400 | 68.7 | 40.7 | 28.9 | 22.4 | 18.2 | 15.4 | 13.1 | 11.5 |
| 500 | 70.6 | 41.4 | 31.2 | 23.8 | 19.3 | 16.2 | 14.1 | 12.3 |
| 600 | 80.4 | 44.5 | 32.8 | 24.7 | 20.4 | 16.9 | 14.9 | 13.0 |
| 700 | 85.0 | 46.0 | 34.6 | 25.7 | 21.3 | 17.6 | 16.1 | 13.8 |
| 800 | 89.5 | 47.2 | 35.7 | 26.3 | 22.0 | 18.1 | 16.2 | 13.9 |
| 900 | 93.0 | 48.2 | 37.1 | 27.0 | 22.6 | 18.5 | 16.6 | 14.4 |
| 1000 | 96.1 | 49 0 | 37.9 | 27.5 | 23.2 | 18.8 | 16.9 | 14.5 |
| 1200 | 102.8 | 50.7 | 40.3 | 28.8 | 24.8 | 19.9 | 17.7 | 15.0 |
| 1400 | 108.6 | 52.0 | 41.5 | 29.3 | 25.9 | 20.5 | 18.6 | 15.7 |
| 1600 | 113.4 | 53.1 | 43.5 | 30.3 | 26.5 | 20.9 | 19.2 | 16.1 |
| 1800 | 117.5 | 54.0 | 44.8 | 31.0 | 27.3 | 21.2 | 19.6 | 16.4 |
| 2000 | 122.0 | 55.0 | 45.8 | 31.4 | 27.5 | 21.5 | 19.9 | 16.5 |

E. F. SCHAEFER.

this value will not be found in practice and 30 per cent. may be assumed as a good practical value for the loss under average conditions. On this basis it is found, in the present case, that to deliver 100 horse-power in compressed air at 100 pounds pressure by one-stage compression, the compressor will require 130 indicated horse-power, ignoring mechanical losses. Looking now at column 4 of the table, the percentage of loss in two-stage compression at this pressure is found to be 17.1 per cent., which is very close to the value which will be found in practice. Applying this value, it is evident that to deliver the equivalent of 100 effective horse-power in air at 100

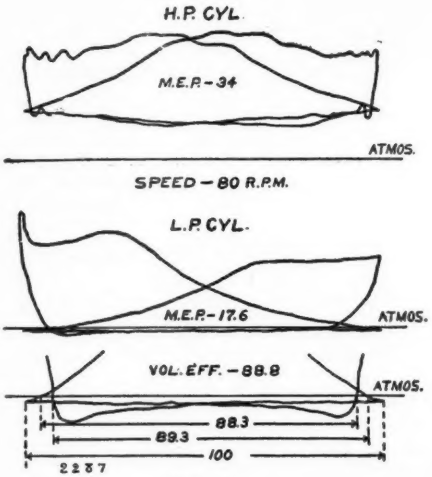



FIG. 5.

pounds pressure by two-stage compression, about 117 indicated horse-power will be required. In this case, as between single and two-stage compression, we have a direct saving in favor of the latter of 13 indicated horse-power or 10 per cent. Considering the compression of the same volume to the same pressure in four stages, the percentage of loss is seen to be 8 per cent. from column 6, implying an applied power of 108 indicated horse-power. In this case the saving, as compared to single-stage compression, is 24 horse-power, or 18½ per cent. From these gains something must be allowed for the friction of extra mechanical parts and of the air through additional sets of ports, valves, coolers, etc. More especially is this true when the machine belongs to that class of machines termed "compound" by courtesy, attractive in price through frugal designing, in which small coolers, insufficient valve area, the use of a hot discharge port for the air intake, small ports, etc., are all antagonistic to economy.

Authentic and repeated tests show that such machines may actually require 10 to 15 per cent. more power per cubic foot of air really delivered than some well designed simple single-cylinder types. No more cylinders are required for the compound than for the simple machine, in

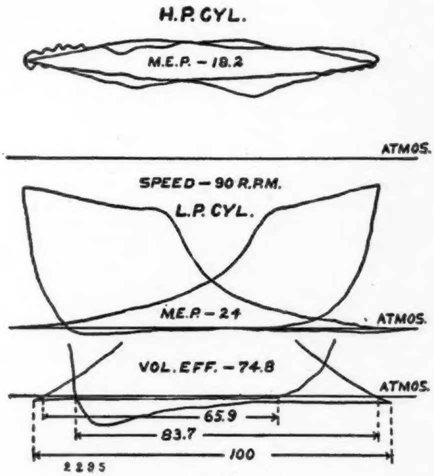



FIG. 6.

duplex constructions. Yet here, too, the economy expected is only realized from high-class designs, generously proportioned, and fitted with large coolers and the other essential refinements of good practice.

*Reduced Strains.*—When compression is carried on in a single cylinder, the difference between the pressures at the beginning and at the end of the stroke is the total difference between initial and terminal pressures, entailing a great variation in strains on the driving mechanism and the structure of the machine. The greatest strains come near the end of the stroke and are almost instantly relieved

when the inlet valves open. Thus the terminal strain on a 20-inch cylinder having 314 square inches area, at 100 pounds pressure will be 31,400 pounds, or nearly 16 tons. At 100 revolutions this strain is repeated 200 times per minute and demands a very rugged construction. This is a condition not conducive to easy operation in any but the most massively proportioned compressors.

In compound compression, on the other hand, the difference between initial and terminal pressures in each cylinder is but a fraction of the total range of pressure. The pressures, furthermore, are partially balanced in the several cylinders. The working strains on valves and other parts are consequently greatly diminished, resulting in a greatly reduced wear and liability to breakage, and securing free lubrication and a noticeable improvement in the way of smooth, easy operation of the machine. These are all facts which contribute to continuous and satisfactory service, with the least possible adjustment and attention. As a matter of fact, compounding the air cylinders transfers so much of the load from the later to the earlier part of the stroke that the maximum terminal strain on bearings is reduced fully 45 per cent. over that in single-stage compression; in the above case, the reduction would be from 3,140 "ton-minutes" to 1,727—obviously a much easier proposition mechanically. Misled by this point, it has been common to reduce the weight and size of bearings accordingly—a mistake which will be evident when it is remembered that the stoppage of circulating water in the cooler at once raises the load on the low-pressure piston; while a broken or damaged outlet valve on the high-pressure cylinder may at any moment throw the same load on all parts as in a single-cylinder machine.

*Improved Steam Economy.*—The more equable distribution of the load throughout the stroke in compound compression, just noted, also aids in securing a higher economy in steam consumption at the lower end of a "straight-line" machine; for it makes possible an earlier cut-off in the steam cylinders and a consequently greater steam expansion with its attendant saving, late cut-offs not being so necessary to prevent "dead-centering." Multi-stage compression with effective inter-coolers between stages also permits a higher piston speed, in itself a factor in steam economy in reducing the leakages and condensation in the steam end.

*Higher Volumetric Efficiency.*—The air remaining in the clearance space at the end of the stroke must be expanded on the return stroke to atmospheric pressure before free air can enter through the inlet valves. Evidently the higher the pressure in the clearance space, the greater will be the expanded volume, and the lower will be the intake efficiency of the cylinder. In single-stage compression the clearance pressure is the working pressure; in compound compression the clearance pressure in each cylinder is the terminal pressure in that cylinder. But this terminal pressure in the intake cylinder is low

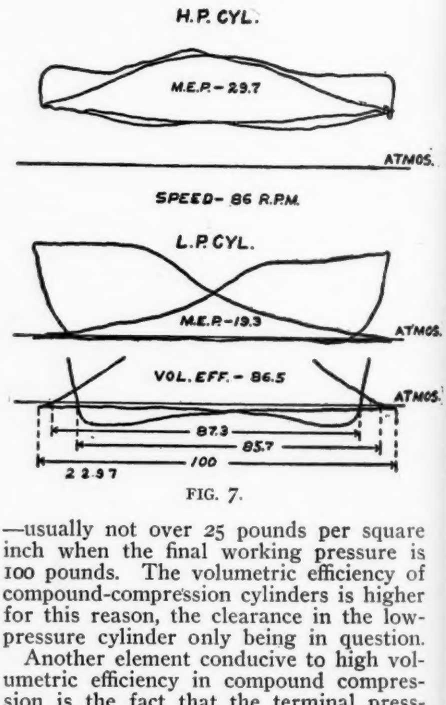



FIG. 7.

—usually not over 25 pounds per square inch when the final working pressure is 100 pounds. The volumetric efficiency of compound-compression cylinders is higher for this reason, the clearance in the low-pressure cylinder only being in question.

Another element conducive to high volumetric efficiency in compound compression is the fact that the terminal pressures, and consequently the terminal temperatures, are lower than in single-stage cylinders. The cylinder walls and, more particularly, the heads, together with the valves and ports which may be in them, are therefore kept much cooler and the entering air is not heated so much by contact with these parts. A third element

entering into the question of efficiency is the reduced leakage through valves and past piston and rods, with the incidental loss of power. It is evident that the higher the pressure the greater the liability to leakage; and the small range of pressures in multi-stage cylinders reduces this loss.

*Dryer Air.*—One of the greatest difficulties hitherto encountered in air-power transmission has been the freezing of the moisture in the air, in the pipe line or at the exhaust ports of the air motors. One of the great advantages of the sub-division of compression into several stages lies in the opportunity it affords for cooling the compressed air at intermediate stages to a temperature at which its moisture will be precipitated. Of course practically all of this condensation occurs in the inter- and aftercoolers; and herein appears a necessity for a design which will pass the air at low velocity with full opportunity for cooling on the water tubes. The moisture in suspension is withdrawn through the drain pipe. It is needless to say that unless some provision is made for arresting and withdrawing the condensed water from the intercooler, the value of the latter as an air-dryer is lost; for the moisture is carried over into the compression cylinders, producing cutting and leakage and working out into the pipe line. Aftercoolers are in some instances as important as intercoolers in removing moisture.

*Better Lubrication.*—If air be compressed in a single cylinder from atmospheric pressure and temperature of 60 degrees Fahr. to a final pressure of 100 pounds, the maximum temperature attained may be 484 degrees Fahr. This temperature is manifestly destructive to common lubricants, and oils of ordinary quality are burned into a solid, gritty, coke-like or gummy substance, which gives the very reverse of proper lubrication. This deposit, moreover, collecting in ports and valves may so obstruct and clog them as to cause leakage and throw an added load on the compressor.

If, however, the same volume of air be compressed in the first cylinder to a pressure of 25 pounds, the highest temperature which can be reached is only 233 degrees—a heat which will not destroy the lubricating qualities of good oils such as should be used in compressor work. This air, passing through the intercooler, will be brought back to about the original temperature of 60 degrees and compressed, in a two-stage compressor, from 25 to 100 pounds in the second cylinder. Here the maximum temperature attained will be but little, if any, in excess of that in the first cylinder, since the heat of compression is a function of the number of compressions, and is almost wholly independent of the initial pressure. In multi-stage compressors, therefore the conditions of temperature are evidently most conducive to thorough lubrication of piston and valves, tending toward durability of working parts, with long life and high efficiency of the machine.

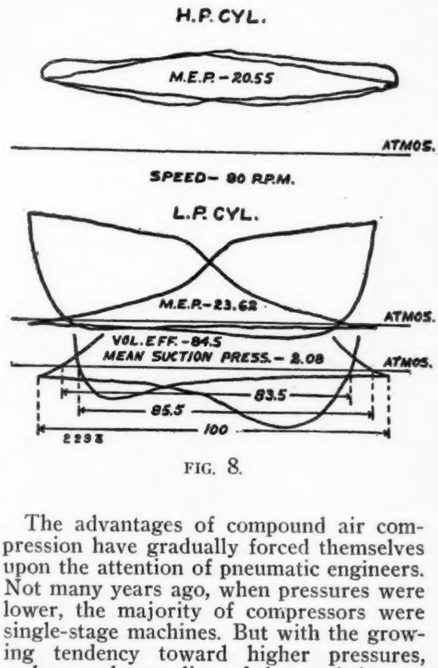



FIG. 8.

The advantages of compound air compression have gradually forced themselves upon the attention of pneumatic engineers. Not many years ago, when pressures were lower, the majority of compressors were single-stage machines. But with the growing tendency toward higher pressures, and an understanding of the needed economies, compound compressors came into greater prominence; and of late much the larger percentage of installations have been machines of this style.

But it will not do to reason that any compound compressor is necessarily more economical than a high-class simple machine, for such is not the case. On the contrary, only compounds of the highest class are advantageous from an econom-

ical standpoint. The gains depend not simply upon stage compression and effective cooling, but also upon correct design throughout the machine and a consistent attention to every detail. Every condition which may possibly affect the air from intake to discharge must be properly considered and provided for. Some of these defects which may offset compression economy have been noted at intervals throughout the preceding discussion. But their importance merits a repetition of them here. A weak structure and insufficient bearings (based on a mistaken idea of reduced strains), with no provision for unexpected contingencies, resulting in excessive friction losses; multiplicity of wearing parts, absorbing a large portion of the power theoretically saved; heated and restricted air passages, inefficient valves, neglect of proper jacket and head cooling; small and ineffective intercoolers; poor workmanship, resulting in leakage losses.

Not only may the enumerated defects largely offset the saving by compression in stages, but it is a fact that compounds now on the market may require more power per cubic foot of air compressed than well-designed, high-class, simple compressors of equivalent capacity. This latter statement embodies so great an apparent engineering heresy as to demand a strong substantiation in fact. For this reason the following test is reported. For obvious reasons the names of the makers of the compressors in question are withheld. It may be stated here, however, that the machines are standard types of American builders of established reputation at home and abroad; the two manufacturers have been distinguished by the terms "Maker A" and "Maker B." The object of the tests was primarily to ascertain the comparative economies of compound compressors of average type and simple machines of the highest grade, more especially under practical conditions of continuous use.

To make the tests fair and representative, compressors in actual service were selected and those tested have been in

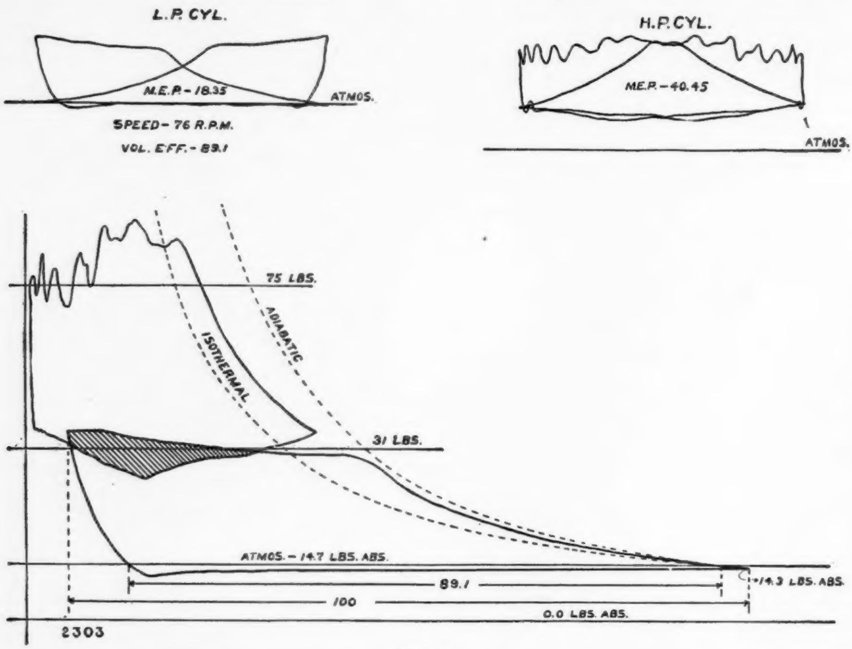



FIG. 9.

TABLE 2. SINGLE-STAGE AIR COMPRESSORS—"MAKER A."

| Machine No. | 1A. | 2A. | 3A. | 4A. | 5A. |
|---|---|---|---|---|---|
| Diameter air cylinder, inches | 26 | 26 | 26 | 26 | 26 |
| Diameter steam cylinder, inches | 24 | 24 | 24 | 24 | 24 |
| Stroke, inches | 30 | 30 | 30 | 30 | 30 |
| Speed, R. P. M. | 88 | 80 | 84 | 60 | 80 |
| Mean net area—air cylinder | 519.8 | 519.8 | 519.8 | 519.8 | 519.8 |
| Air cylinder displacement | 1,590 | 1,445 | 1,518 | 1,080 | 1,445 |
| Free air shown by L. P. card | 1,383 | 1,285 | 1,324 | 973 | 1,250 |
| Volumetric efficiency, % | 87 | 89.1 | 87.4 | 91 | 86.5 |
| Receiver pressure | 61 | 60 | 61 | 92.5 | 95 |
| Horse-power, Air cylinder | 201.5 | 185 | 192 | 171 | 223.5 |
| Air horse power per 100 cu. ft. free air | 14.51 | 14.4 | 14.5 | 17.6 | 17.86 |
| Computed horse-power per 100 cu. ft. adiabatic comp. | 13.5 | 13.35 | 13.5 | 17.3 | 17.52 |
| Efficiency compared with perfect one-stage comp., % | 93 | 92.8 | 93.2 | 98.4 | 98.2 |
| Computed horse-power per 100 cu. ft. perfect two-stage comp. | 11.9 | 11.8 | 11.9 | 14.83 | 15.05 |
| Efficiency compared with perfect two-stage comp., % | 82 | 82 | 82.1 | 84.5 | 84.3 |
| Indicator Card | Fig. 1 | Fig. 1 | | | |

TABLE 3. TWO-STAGE AIR COMPRESSORS—"MAKER A."

| Machine No. | 6A. | 7A. | 8A. | 9A. |
|---|---|---|---|---|
| Diameter L. P. air cylinder. inches | 20 | 20 | 15 | 26 |
| Diameter H. P. air cylinder, inches | 13 | 13 | 10 | 16 |
| Diameter steam cylinder, inches | 18 | 18 | 14 | 24 |
| Stroke, inches | 18 | 18 | 14 | 24 |
| Speed, R. P. M. | 100 | 80 | 155 | 108 |
| Mean net area L. P. air cylinder | 308 | 308 | 171.33 | 515 |
| Mean net area H. P. air cylinder | 132.46 | 132.46 | 78.54 | 198.45 |
| L. P. cylinder displacement | 642 | 513 | 430 | 1,545 |
| Free air shown by L. P. card | 616 | 495 | 398 | 1,448 |
| Volumetric efficiency, % | 96 | 96.4 | 92.6 | 93.8 |
| Intercooler pressure | 23 | 23 | 22 | 25 |
| Receiver pressure | 85 | 82 | 90 | 90 |
| Horse-power L. P. air cylinder | 44 | 35 | 28 | 106 |
| Horse-power H. P. air cylinder | 52.25 | 40.1 | 35.8 | 122.8 |
| Total air horse-power | 96.25 | 75.1 | 63.8 | 228.8 |
| Air horse-power per 100 cu. ft. free air | 15.6 | 15.15 | 16 | 15.82 |
| Computed horse-power per 100 cu. ft. two-stage comp | 14.22 | 13.92 | 14.71 | 14.7 |
| Efficiency compared with perfect two-stage comp., % | 91.25 | 92 | 92 | 93 |
| Indicator Cards | Fig. 2 | | | Fig. 3 |

TABLE 4. TWO-STAGE AIR COMPRESSORS—"MAKER B."

| Machine No. | 1B. | 2B. | 3B. | 4B. | 5B. |
|---|---|---|---|---|---|
| Diameter L. P. air cylinder, inches | 28 | 28 | 28 | 28 | 26 |
| Diameter H. P. air cylinder, inches | 17½ | 17½ | 17½ | 17½ | 17½ |
| Diameter steam cylinder, inches | 28 | 28 | 28 | 28 | 24 |
| Stroke, inches | 30 | 30 | 30 | 30 | 30 |
| Speed, R. P. M. | 76 | 80 | 90 | 86 | 80 |
| Mean net area L. P. air cylinder | 611.9 | 611.9 | 611.9 | 611.9 | 527.4 |
| Mean net area H. P. air cylinder | 231.9 | 231.9 | 231.9 | 231.9 | 231.2 |
| L. P. cylinder displacement | 1,618 | 1,700 | 1,915 | 1,830 | 1,465 |
| Free air shown by L. P. card | 1,440 | 1,510 | 1,430 | 1,580 | 1,240 |
| Volumetric efficiency, % | 89.1 | 88.8 | 74.8 | 86.5 | 84.6 |
| Intercooler pressure | 31 | 28 | 50 | 33 | 34 |
| Receiver pressure | 75 | 64 | 68 | 58 | 55 |
| Horse-power L. P. air cylinder | 129 | 130.8 | 200 | 154 | 151 |
| Horse-power H. P. air cylinder | 108 | 95.4 | 57.5 | 89.5 | 57.5 |
| Total air horse-power | 237 | 226.2 | 257.5 | 243.5 | 208.5 |
| Air horse-power per 100 cu. ft. free air | 16.45 | 15 | 18 | 15.4 | 16.82 |
| Computed horse-power per 100 cu. ft. two-stage comp | 13.3 | 12.28 | 12.7 | 11.62 | 11.27 |
| Efficiency compared with perfect two-stage comp., % | 80.9 | 81.8 | 70.5 | 75.5 | 67 |
| Indicator Cards | Fig. 4<br>Fig. 9 | Fig. 5 | Fig. 6 | Fig. 7 | Fig. 8 |

operation for from two to fourteen years under average working conditions and average attendance. But the results showed up the compounds in such an unfavorable light, that it was determined to continue the investigation, to include compounds of the same grade of workmanship and the same make as the simple machine tested. However, no old compressors of this type were available, so new machines were tested at the builder's shops, just before shipment. The conditions in all tests were made as nearly identical as possible and the results show in a striking manner the effect of correct design and workmanship on compressor efficiency.

The thing which appeals to the buyer is the economy of the machine he purchases —the saving he will secure by its use. In an air compressor, two distinct efficiencies are to be considered; the volumetric efficiency, or ratio of actual to theoretical air capacity; and the compression efficiency, or ratio of actual power required to the theoretical. To facilitate comparison in the present case, the several values from the preceding report are here grouped and averaged:

VOLUMETRIC EFFICIENCIES.

"Maker A," two-stage—96, 96.4, 92.6 and 93.8 per cent.; average, 94.7.
"Maker A," single-stage—87, 89.1, 87.4, 91 and 86.5 per cent.; average, 88.2.
"Maker B," two-stage—89.1, 88.8, 74.8, 86.5 and 84.6 per cent.; average, 84.76.

It here appears that the high-grade two-stage compressors of "Maker A" suffer an average loss in capacity of only 5.3 per cent. of the theoretical; the high-grade single-stage machines lose on the average only 11.8 per cent. of the theoretical capacity; while the ordinary two-stage compressors of "Maker B" lose an average of 15.24 per cent. of their theoretical air displacement. High-grade workmanship and correct design, therefore, effect in these instances an average saving in air capacity of 9.94 per cent. in the two-stage types and 3.44 per cent. in single-stage machines.

COMPRESSION EFFICIENCIES.

"Maker A," two-stage—91.25, 92, 92 and 93 per cent.; average, 92.06.
"Maker A," single-stage—82, 82, 82.1, 84.5 and 84.3 per cent.; average, 82.98.
"Maker B," two-stage—80.9, 81.8, 70.5, 75.5 and 67 per cent.; average, 75.14.

The high-grade compounds waste but 7.94 per cent. of the theoretical power required; the single-stage machines of the same grade lose 17.02 per cent.; while the ordinary two-stage compressors of "Maker B" waste 24.86 per cent. of the power applied to them.

The striking feature of this is the enormous waste in the compounds of "Maker B," due to defects such as have been briefly suggested. These losses have more than offset the advantages of compounding. Even the single-stage machines of "Maker A" show an average superiority of 7.84 per cent. over these defective compounds, while comparison of the two two-stage types shows an average advantage of 16.92 per cent. in favor of "Maker A's" high-class machines, correctly designed and properly proportioned for the duty.

Compression efficiencies for "Maker A's" single-stage machines, in the last line of the table, are referred to perfect two-stage compression, simply for ready comparison with the compound machines. It is to be noted that the efficiency of these compressors compared to perfect one-stage compression is also given as follows: 93, 92.8, 93.2, 98.4 and 98.2 per cent.; average, 95.12. This means that on the average these machines depart less than 5 per cent. from theoretically perfect performance in their type. Such efficiencies indicate a remarkable refinement in design and workmanship and exemplify the importance of the care bestowed upon what are usually considered minor details.

The term "compound" or "two-stage" as applied to air compressors should properly stand for superior economy. The buyer of a compound rightfully expects a saving by its use. The foregoing tests, taken without prejudice from average machines in ordinary practice, indicate how even reputable builders, acting no doubt in all good faith, may furnish a compressor which may prove a veritable "gold brick" in the hands of the trusting purchaser. Poor practice may prove the undoing of the best theory. That compressor only is a commercial and economical success which embodies a sound theory in a mechanical structure correctly designed, built by skilled and careful workmen, and so simple as to be readily understood, handled and maintained by mechanics of average intelligence.

## Wet vs. Dry Compressors.

BY J. WALTER PEARSE.

So early as 1867 the late Professor Mallard, of the Paris School of Mines, pointed out that water in compressing cylinders causes a notable diminution of the energy to be expended, and a considerable lowering of the air temperature. This temperature, while being scarcely 50 degrees cent. (122 Fahr.) in wet compressors, attains as much as four times that figure in the dry variety, which is one of their inherent disadvantages, alone sufficing to determine the adoption of compression in stages. The Professor also showed that the quantity of water to be introduced is slight, but that there is no great disadvantage in employing a slight excess of water, the quantity of which can easily be determined for each compressor.

It may be added (observed M. J. Francois, at the Liege Mining Congress) that the introduction of water into the cylinders requires the use of valves with hydraulic joints, which are far more efficient than those with dry joints. To have a just idea of this, it is necessary to determine the relative speed of the water and compressed air, which flows off freely through one and the same orifice, and also to consider that the volume of the compressed air leakage must be multiplied by $\frac{P_1}{P_0}$ ($P_0$ and $P_1$ representing the respective pressures of the air on entering and leaving the compressor) at any rate, for the leakage which occurs during aspiration, this being necessary for determining the influence of the leakage, not only on the volumetric yield, but also on the dynamic yield, seeing that in this case the loss is of compressed air.

Water introduced into compressors, provided it be recuperated and used over again, after being passed through a refrigerator along the return pipe, becomes absolutely neutral and incapable of causing injurious incrustations on valves, pistons and cylinders, even if the water at disposal be not of the best quality. Moreover, if a little soft soap be added to the water it will, of itself, be sufficient to lubricate the pistons and cylinders.

Nor should it be forgotten that the water is drawn along in the state of steam through the pipes leading underground, and, owing to the fact that such steam is produced by the heat disengaged from the air during its compression, the quantity of water thus lost will always be very slight. From the heat due to the compression must be deducted that lost in the water recovered, as also the loss owing to the cooling action of the compressing cylinders.

The author brings figures to show that in a compressor of 100 indicated horsepower, even supposing that all the heat engendered by the compression vaporizes the water, less than half a litre (or not a pint) of water is thus lost per horsepower per hour, while this very slight quantity is a maximum which is far from being actually attained.

The disadvantage sometimes attributed to dampness in compressed air, as regards its freezing in the motors, is negligible, because it may be avoided by previously determining the space to be left free between piston and cylinder cover, and so regulating the distribution of compressed air as to obtain the desired compression in this space. Such practice, the advisability of which may be proved mathematically, prevents the formation of hoar-frost inside the motors, and it causes no loss of power, because that absorbed in compression is at once given back.

In any case, whether the compressor be wet or dry, M. Francois advocates making the delivery passages as short as possible, and only employing for cylinder lubrication such oils as are used in ice-making machines, while in large motors the compressed air should be delivered into a wooden "snow-box" by pipes leading sufficiently far from the motor to preserve it and the driver from an icy cold.

His conclusions are that, although air compression can be effected by both dry and wet compressors, the latter have great advantages, both in single stage and multiple compression. In point of fact, if a comparison be made between the two methods now almost exclusively employed—multiple compression in dry compressors and direct compression in wet compressors—it will be found that the former has a certain advantage as regards the dynamic yield, although the extent of this advantage has been somewhat exaggerated.

The author's examination of the re-

sults shown by indicator diagrams in the two methods of compression, up to 6 and also up to 7 atmospheres, shows that in the first case, for the compression of a cubic metre of natural air, a work of 26,050 kilogram metres is required by direct compression in a wet compressor, of 24,000 kilogram metres by dry compound compressors, showing an advantage of 7.8 per cent. in favor of the latter, while in the second case (compression to 7 atmospheres), the economy will be 28,300 — 25,925 — 2,375 kilogram metres, or 8.7 per cent.

Accordingly, this is the only advantage of dry multiple compression over wet direct compression for pressures not exceeding those adopted in mines, and it must be added that this advantage is set off by the complication of the two stages.

## Air Compressor at the Champion Copper Mine, Painesdale, Michigan.

The mining of copper is probably the most expensive form of mineral production; and of all copper mines, those of the "copper country" of Lake Superior have the highest operating cost. Low-grade values, extended deposits, and immense depths make mining costs high and necessitate the most economical methods of operation to keep these properties in the favored list of dividend payers.

Among the twenty-two producing mines of the Lake Superior district the "Champion" ranked fifth in production in 1904, being credited during that year with an output of 12,500,000 pounds of refined copper, the sale of which yielded dividends of $300,000. The mine is at Painesdale, Mich., and is one of several large properties owned and operated by the Copper Range Consolidated Copper Company. Under present average conditions of operation its output is about 2,000 tons of ore per day, which is hoisted through four shafts well scattered over the extensive property.

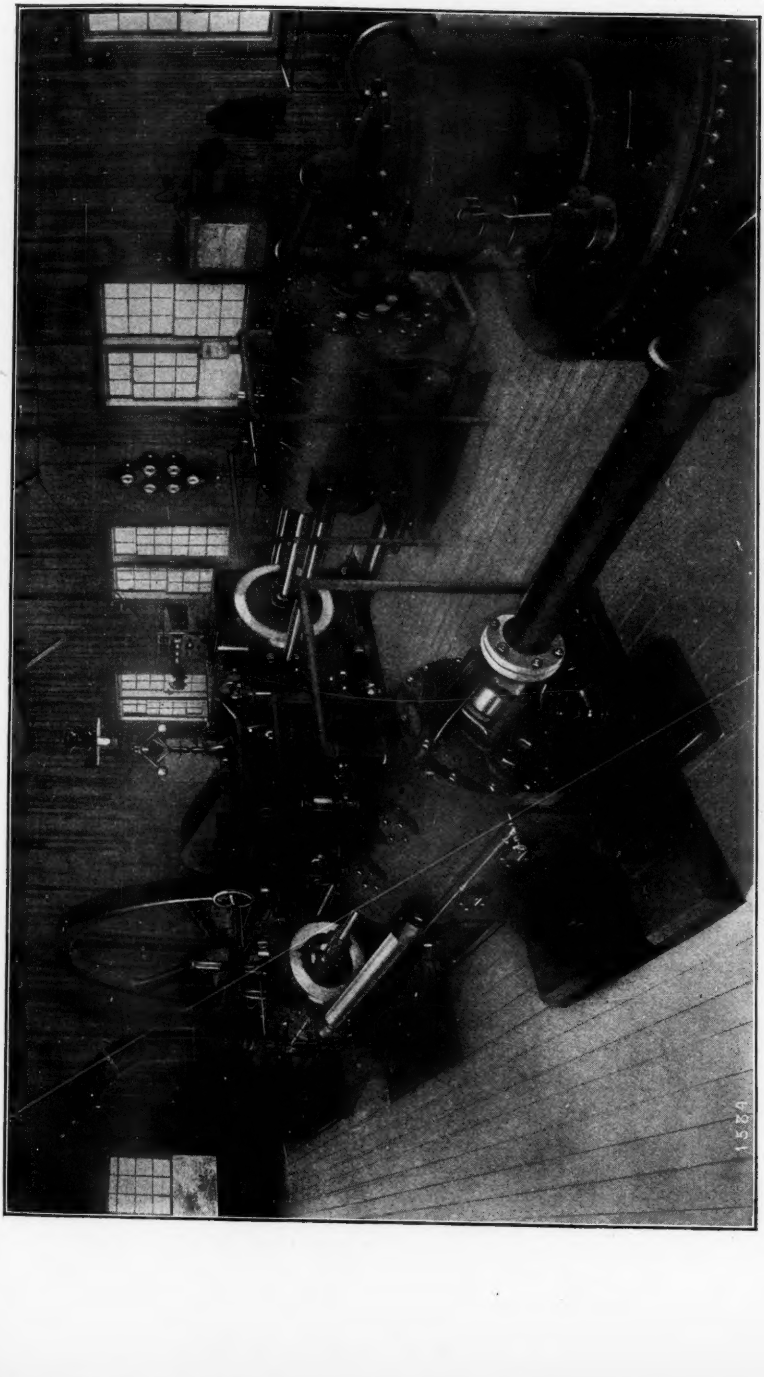
High-grade machinery equipments are characteristic of Michigan copper mines, and the "Champion" is no exception. Cheap power is but one factor in securing the low operating cost made necessary by mining conditions. Machine methods rule and the rock drill is monarch of the situation. The air power plants of these mines are the key to their successful operation, and that of the "Champion" is typical of the high standard of economy maintained.

One of the largest compressors in the district is located on this property and was built by the Ingersoll-Sergeant Company, of New York. It is a duplex Corliss type with cross-compound condensing steam cylinders and two-stage air cylinders. Steam is supplied at 130 pounds pressure by a battery of high pressure boilers in an adjacent boiler house. The steam pipe enters the basement of the compressor house and rises through the engine room floor to the high-pressure steam cylinder with a wide sweep bend of 6-inch pipe. The exhaust passes to a Worthington condenser, located, with its pumps, in the basement below. A vacuum of about 20 inches is maintained. The steam-valve gear is of standard Corliss release type. Automatic regulation is secured by an improved governor, in which the usual fly-balls are supplemented by a governor cylinder through which receiver air pressure operates to vary the cut-off on both cylinders, adjusting the speed to the demand for air and maintaining air pressure constant. Main frames are of standard heavy-duty tangye pattern, and cranks are of balanced disk type.

High and low-pressure air cylinders are completely water-jacketed on heads and barrels. Cold air is drawn from outside the building through a duct terminating in an intake bracket supported on the compressor foundation. The Sergeant piston inlet valve handles the intake on both high and low-pressure cylinders. Direct-lift discharge valves are used. A vertical receiver intercooler stands between the cylinders, 48 inches in diameter and 13½ feet high, with tinned brass tubes through which the cooling water circulates.

Steam cylinders are 18 and 36 inches in diameter; air cylinders, 20¼ and 32¼ inches. The stroke is 48 inches. Normal speed is 72 revolutions per minute, at which the free air capacity of the compressor is 3,100 cubic feet per minute. Air pressure is 80 pounds.

The compressor building is an iron-covered steel-framed structure lined with sheathing and floored with matched wood. A solid concrete foundation capped with



AIR COMPRESSORS IN THE POWER HOUSE AT PAINESDALE, MICH.

finished stone supports the compressor. The illustration shows the interior of the engine-room with the intercooler in the right foreground. The stairway on the right leads to the basement, which contains all auxiliaries, beside an engine-driven generator for lighting purposes.

In a separate building, close at hand, is another Ingersoll-Sergeant air compressor—a survivor from early days of the "Champion" mine. At one time this small machine furnished all the air required for rock drills; to-day it is maintained as an auxiliary to the larger compressor. It is a straight-line type, steam-driven, with simple steam and air cylinders, 18 and 18¼ inches in diameter, respectively, and a stroke of 24 inches. At the rated speed of 94 revolutions per minute its free air capacity is 656 cubic feet per minute, delivered at 80 pounds pressure. The steam end is fitted with adjustable cut-off valves of Meyer type; and Sergeant "piston inlet" and direct-lift discharge valves handle admission and discharge of air.

There was a period of two years in the history of the "Champion" mine when rapid development threw an unusually heavy load on these two compressors—the Corliss and straight-line. During this time these machines ran continuously, 24 hours per day and 7 days in the week, carrying, jointly, an average load of 56 rock drills, 7 pumps and 4 "puffers"—small hoists used for light work. Throughout all this heavy overload there were no breakages and no repairs on the compressors, nor any delay directly chargeable to these machines. They came through the ordeal in splendid condition. Now that conditions are somewhat relieved the larger compressor easily does the work of its section of the mine, while its smaller associate stands ready for emergencies.

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## Air Required to Operate Rock Drills.*

The first thing to consider when about to install compressed air to operate rock drills is the number and sizes of the drills required to do the work. This means the total number of drills that will be in active service at one time.

* **Written for *Mines and Minerals*, by F. M. Hitchcock, M. E.**

The size of the drills is determined by the material to be drilled, whether soft, medium, or hard, together with the depth that it is desired to drill the holes.

The number of drills to use is decided by the rapidity with which the work is to be done, the feasibility of working more or less drills at different points to advantage, etc.

When the maximum number to be run is determined, the question arises as to what capacity of compressor is needed to drive them. This depends greatly on the air pressure to be used, which varies from 60 pounds to 100 pounds in different plants; the lower the air pressure, the less the amount of air required. For use at sea level, probably 80 pounds is the most common pressure at the present time. This seems to be the pressure that gives the best results under average conditions, but the kind of rock to be drilled, and the hardness of the drill steel, are the deciding factors. Few drills are worked economically with air at a pressure above 100 pounds, on account of the rapidity with which the steel dulls, and the amount of vibration, particularly in hard rock. The frequent sharpening and changing of bits greatly increases the loss of time.

The accompanying table 1 shows the amount of air required to run from one to 50 Rand drills at an air pressure of 80 pounds at sea level. It is computed on a sliding scale, determined by experience, and gives the air required under usual working conditions. This means that with the increase in the number of drills used, less air is required per drill, or that there is a decrease in the percentage of the number that will be in simultaneous operation.

The amount of air that a compressor should furnish to run a certain number of drills at 80 pounds pressure may at once be determined from the table.

To find the air required for other pressure than 80 pounds multiply the amount given in the table by the following factors:

For 60 pounds by .789. For 70 pounds by .894. For 90 pounds by 1.105. For 100 pounds by 1.211.

The quantity determined, however, is the free air that a compressor should *deliver* compressed, and not its displacement capacity.

The figures given do not take into consideration the loss of pressure that the air sustains in its transmission from the

compressor to the drills, due to friction and leakage in the pipes. In a properly constructed line, however, these losses do not amount to a great deal, and air is carried 5,000 to 10,000 feet with a loss of only 3 to 5 pounds in pressure.

TABLE 1—*Air Required to Operate Rand Rock Drills at Sea Level, With Air at 80 Pounds Pressure Per Square Inch.*

| Name of Drill | Kid | No.1 | No.2 | No.3 | No. 3¼ | No.4 | No.5 | No.7 |
|---|---|---|---|---|---|---|---|---|
| Diameter of Cylinder, in Inches | 1⅞ | 2¼ | 2¾ | 3⅛ | 3¼ | 3⅝ | 4½ | 5½ |
| No. of Drills | | | | | | | | |
| 1 | 44 | 67 | 81 | 120 | 130 | 141 | 166 | 194 |
| 2 | 77 | 117 | 142 | 211 | 228 | 247 | 292 | 340 |
| 3 | 114 | 173 | 210 | 310 | 336 | 364 | 428 | 503 |
| 4 | 150 | 228 | 275 | 408 | 442 | 478 | 564 | 659 |
| 5 | 183 | 279 | 337 | 500 | 542 | 587 | 692 | 810 |
| 6 | 216 | 329 | 397 | 588 | 638 | 692 | 814 | 952 |
| 7 | 246 | 375 | 454 | 673 | 728 | 790 | 928 | 1,085 |
| 8 | 275 | 418 | 507 | 750 | 813 | 882 | 1,040 | 1.212 |
| 9 | 301 | 458 | 554 | 820 | 890 | 964 | 1,135 | 1,327 |
| 10 | 324 | 494 | 597 | 884 | 957 | 1,040 | 1,222 | 1,430 |
| 12 | 376 | 573 | 692 | 1,025 | 1,110 | 1,205 | 1,418 | 1,658 |
| 15 | 452 | 688 | 830 | 1,230 | 1,334 | 1,446 | 1,705 | 1,990 |
| 20 | 602 | 915 | 1,105 | 1,640 | 1,775 | 1,925 | 2,263 | 2,650 |
| 25 | 748 | 1,140 | 1,378 | 2,043 | 2,218 | 2,400 | 2,820 | 3,300 |
| 30 | 898 | 1,368 | 1,655 | 2,455 | 2,658 | 2,890 | 3,380 | 3,960 |
| 40 | 1,195 | 1,820 | 2,220 | 3,260 | 3,535 | 3,830 | 4,520 | 5,270 |
| 50 | 1,495 | 2,275 | 2,750 | 4,060 | 4,420 | 4,790 | 5.640 | 6,600 |

As the air decreases in density with the height above sea level, more free air must be compressed at altitudes than at sea level to give equivalent results.

Having noted in Table 1 the amount of air required to run a certain number of drills at 80 pounds pressure sea level, multiplying that amount by the factors in Table 2 will give the amount which is equivalent in effect at various altitudes, either at 80 pounds, or at 60, 70, 90 or 100 pounds pressure.

For an example, assume that we wish to run six No. 3 drills at 10,000 feet altitude, and at 90 pounds pressure. From Table 1 we see that they require 588 cubic feet of free air at sea level when running with air at 80 pounds pressure. From Table 2 we find the factor for the conditions given to be 1.542; hence, the amount of air required is $588 \times 1.542 = 906$ cubic feet.

TABLE 2—*Factors for Altitudes.*

| Altitude in Feet Above Sea Level | Atmospheric Pressure Pounds Per Square Inch | Factors | | | | |
|---|---|---|---|---|---|---|
| | | Air Pressure at Drill | | | | |
| | | 60 Lbs. | 70 Lbs. | 80 Lbs. | 90 Lbs. | 100 Lbs. |
| Sea Level | 14.75 | .789 | .894 | 1.000 | 1.105 | 1.211 |
| 1,000 | 14.20 | .814 | .923 | 1.033 | 1.142 | 1.252 |
| 2,000 | 13.67 | .839 | .952 | 1.066 | 1.179 | 1.293 |
| 3,000 | 13.16 | .865 | .983 | 1.102 | 1.220 | 1.338 |
| 4,000 | 12.67 | .893 | 1.016 | 1.139 | 1.261 | 1.383 |
| 5,000 | 12.20 | .922 | 1.049 | 1.176 | 1.303 | 1.431 |
| 6,000 | 11.78 | .952 | 1.085 | 1.218 | 1.350 | 1.482 |
| 7,000 | 11.30 | .982 | 1.119 | 1.257 | 1.395 | 1.534 |
| 8,000 | 10.87 | 1.015 | 1.157 | 1.300 | 1.443 | 1.586 |
| 9,000 | 10.46 | 1.049 | 1.198 | 1.347 | 1.494 | 1.642 |
| 10,000 | 10.07 | 1.083 | 1.236 | 1.390 | 1.542 | 1.695 |
| 11,000 | 9.70 | 1.119 | 1.279 | 1.439 | 1.599 | 1.760 |
| 12,000 | 9.34 | 1.157 | 1.322 | 1.488 | 1.656 | 1.824 |
| 13,000 | 8.98 | 1.197 | 1.370 | 1.543 | 1.716 | 1.889 |
| 14,000 | 8.65 | 1.236 | 1.415 | 1.594 | 1.775 | 1.956 |
| 15,000 | 8.32 | 1.279 | 1.465 | 1.652 | 1.840 | 2.028 |

## Notes.

POSTMASTER-GENERAL CORTELYOU has decided to endorse recommendations for the extension of the pneumatic mail-tube service in Brooklyn.

ALFRED E. SHARPE, a fourteen-year-old boy of Iowa Falls, Iowa, has just completed an engine which may be run by either steam or compressed air.

THE Ohio Compressed Air House-cleaning Co., of Columbus, has been formed with the following incorporators: J. N. Koerner, E. J. Burkett, Walter L. Lillie, John L. Trauger and F. M. Leonard.

As a rule coal cutters of the compressed air puncher-type are more effective than electric cutters in low coal workings containing clay spots, which are much faulted. Rotary coal cutters are best suited to thick seams of clean coal.—*Engineering and Mining Journal.*

LORD MASHAM (Samuel Cunliffe-Lister). the patentee of many inventions, including a compressed air brake for railroads and a wool combing machine, died February 2, at Swinton Abbey, Masham, County of York, England. He was born in 1815 and was created a baron in 1891.

In a shop where the sand blast is now being used for cleaning crown bars and crown sheets, it formerly took a 17½-cent man 10 hours to clean a dozen bars —$1.75 for the lot. With the sand blast, a bar is cleaned in 20 or 30 minutes, or about half the time it took using the old way.—*Boiler Maker.*

Compressed air is sometimes used as a temporary relief in rooms in coal mines when the fan currents are insufficient. It is not an advisable proceeding, and is evidence of bad management. Every coal mine should have a fan capacity ample for all calls made upon it.—*Engineering and Mining Journal.*

Power may often be transmitted from the surface to the face of the workings in a coal mine by compressed air better than any other means. The chief disadvantages of the compressed-air system are, danger of dislocation of the pipe lines through fall of roof, and less flexibility when applied to underground haulage than the electric system.—*Engineering and Mining Journal.*

The Westinghouse Machine Company has opened a Philadelphia sales office in room 1,003, North American building. The establishment of this office was necessitated by the rapidly expanding business in this territory, particularly in gas engines and Westinghouse-Parsons steam turbines, and is in line with the progressive policy of the company to establish headquarters in all large industrial cities.

The term "capacity," when applied to an air compressor, is often misunderstood. It relates to the piston displacement in the large low-pressure cylinder of the compressor; it does not refer to the amount of compressed air the machine will deliver. The same machine when working under different pressures, or at different elevations above sea level, has the same capacity; but delivers different quantities of compressed air per unit of power.—*Engineering and Mining Journal.*

A high-altitude air compressor is designed to compress the lighter air in high altitudes to the same gauge pressure that another machine produces when working at sea level. At any appreciable altitude above sea level a larger number of compressions are required to produce the same sea-level gauge pressure. This is arranged by increasing the area of the air cylinder in the high-altitude compressor, and the valves are set so as to effect an earlier cut-off.—*Engineering and Mining Journal.*

An engineering and machinery exhibition will be held in London, England, from September 15 to October 17, 1906, under the presidency of Sir Wm. H. White, K. C. B., LL. D., F. R. S., and a distinguished patronage, including the foremost English scientists; the past presidents, presidents and vice-presidents of all the engineering institutions, the consuls of foreign countries, etc. While this exhibition has been held annually for some time, it has been on a much smaller scale than is planned for this year's.

There will be a full and generously illustrated description of the "New New York Custom-house" in the March *Century* by Mr. Charles de Kay. The fine new structure now nearing completion stands on Bowling Green, Whitehall Street and Battery Park, seven stories from street to roof, with two stories below the street. Convenient approaches, elevators, sunlight, electricity and ventilation will be conspicuous features of the new Custom-house as well as notable architecture and sculpture.

Gasoline air-compressors may often be employed advantageously in work that does not warrant the installation of a steam plant. A 3¼-inch drill will require about 12 horse-power, or 12 pints (1.5 gallons) of gasoline, which, at price of 15 cents per gallon, would make the operation for eight hours come to $1.20, reckoning that the drill is run two-thirds time. There may be also saving in attendance, as compared with a small steam plant. Gasoline air-compressors have been successfully employed in mining work.—*Engineering and Mining Journal*

The Cyclone Drill Co., Orrville, O., has been incorporated January 22, with a capital of $100,000, buying out the assets, including machinery, merchandise, patents, etc., of the Cyclone Drilling Machinery Co. and the Preslar-Crawley Mfg. Co. The new company will make drills of all

types and also a combination drill embodying the hollow rod, cable and core drills. The directors of the company are: B. S. Cope, president; Chris. Smith, vice-president; Wm. H. Tschantz, treasurer; Wm. M. Knoppes, secretary, and Walter A. Knight.

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PNEUMATIC tools have recently received an unexpected but deserved mark of approval by the agreement of the Workmen's Society with the Federated Shipbuilding & Engineering Employers of England and Scotland, to encourage the use of these appliances and to allow a substantial reduction in rates of pay for work done by their aid. Hereafter men using pneumatic tools will have their pay reduced 40 per cent. on shell plating and 35 per cent. on interior work generally. Such a concession would manifestly not be made unless the workmen were certain by experience that it was just to both hand and pneumatic work.—*Engineers' Record.*

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A MACHINE for the manufacture of liquid air, the third to be installed in American colleges, is now a part of the equipment in the department of physics at the University of Minnesota. The machine is being used for experiments in radio-activity.

Liquid air can be manufactured in half an hour with the machine. The process is the usual one. After being freed of impurities the air is passed through a compressor such as is used on torpedo-boats for firing torpedoes. It is then passed through potassium hydroxide to condense the gases. After liquefying and expansion processes are completed the air is passed through tubes and collected.

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A CORRESPONDENT of the "Wandsworth Borough News" has just brought to notice some interesting trials which took place nearly sixty years ago with a compressed air motor-car constructed in accordance with the patent of Baron Anthony Bernard Von Rathen, bearing date 2d November, 1847 (No. 11932). The machine was built at the workshops of that ill-fated institution, the Putney College of Civil Engineering, and a public trial of the car took place on the road between the college and Wandsworth on the 9th August, 1849, and again on the 11th of the same month. The car attained a speed of eight miles per hour on the first occasion and twelve miles at the second trial. The inventor seems to have displayed much ingenuity in the details of his machine, and it is somewhat surprising that nothing more was heard of it.—*Motor Car Journal* (Eng.).

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FREDERICK T. TOWNE, superintendent of the Yale & Towne Mfg. Co.'s plant at Stamford, Conn., and son of Henry R. Towne, president of the company, died early on the morning of Sunday, February 4th.

Although only 34 years of age, Mr. Towne had made a name for himself in the industrial world. He was a member of the administrative council of the National Founders' Association from February, 1900, until November, 1903, during the last year of which time he served as president, although he was one of its youngest members. He was exceedingly active in its behalf and had much to do with formulating its policies.

Mr. Towne was a graduate of Tufts College. After completing his course there he entered the employ of the Yale & Towne Mfg. Co., speedily working himself up to the superintendency of its plants at Stamford and Branford, Conn., with 3,000 men in his control. It was his ambition to make the Stamford plant a model one, especially in the relations of employer to employes. He was a member of the American Society of Mechanical Engineers.

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RIDGELY & JOHNSON Tool Company, of Illinois, control a patent recently granted to Henry H. Vaughan, of Cleveland, Ohio, on an improvement in pneumatic hoists.

The invention relates in general to fluid-pressure-operated lifted apparatus, and more particularly to pneumatic hoists. In a fluid-pressure-operated mechanism in which a pull or push is exerted by fluid-pressure medium, it is desirable that the mechanism should be retained in position to maintain the push or pull after the fluid pressure has been disconnected from the motor cylinder, thereby preventing condensation of steam or from leakage of compressed air. It is also desirable that the mechanism should be retained in position to maintain the push or pull until the load has either been removed or has been lifted by the re-ad-

mission of fluid pressure to the motor cylinder. The invention consists in a motor cylinder, a piston fitting within the cylinder, automatic means for retaining the piston in any position to which it may be moved by fluid pressure, and means for releasing the retaining means only when the load is no longer sustained thereby.—*The Manufacturer.*

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AN interesting feature of mining in the Cripple Creek district, and an item which is as closely watched by the operator as any other, is the cost of making air per drill shift, upon which largely depends the possibilities of economical operation. The wide range in this one respect is shown by the fact that in some instances the cost is as high as $3.00 per drill shift, while in others the same results are obtained at a cost of 53 cents. This great difference is due to the class of machinery used and the care and intelligence with which it is operated. Of the different records obtained from mine operators the following seems to show the greatest economy.

Mr. John Sharp, the well known lessee who is operating the Morning Glory of the Work, and the Colorado Boss of the Cripple Creek Consolidated Company, furnished data covering a period from September 20, 1904, to January 17, 1905, during which time 1,367 drill shifts were operated. As shown by his books during that time the coal bills amounted to $1,183.17. The greatest number of shifts worked in one month was in October, when 492 were employed.

The rock hoisted amounted to 7,500 tons during the months of October, November and December, and the coal bills for hoisting amounted to $450.00, leaving the total coal bill for running the air compressor $733.17, or 53.7 cents per drill shift.—*Mining Reporter.*

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AN interesting example of the general utility and adaptability of the Westinghouse air brake pump was given in connection with the raising of the lake steamer "Corey" exclusively by the use of twenty of these pumps. The "Corey" went aground on the shores of Gull Island, Lake Superior, during the severe gale of November 28 last. On the first of December, men and pumps were sent by two of the prominent northern railroads to prosecute the work of raising this vessel.

Ten 11-inch pumps and ten 9½-inch pumps were distributed along the deck of the steamer and arranged to force air into thirteen of the air-tight compartments in the vessel. This work was commenced on the second of December, and while the air was forced into the compartments by the pumps, the leaks were repaired by boilermakers as fast as they were discovered, either by being patched or by the use of cement. To accomplish this preliminary work of stopping all leaks required until December 9, about 3 P. M. At this time all of the pumps were set to work forcing air into the compartments in order to displace the water and raise the vessel. On the following morning at 7:30 A. M., it was found that the boat had been raised four feet and sufficient water removed by the compressed air to allow the steamer "Houghton" to easily pull the "Corey" from the shore into deep water. In raising this vessel it is estimated that from 3 P. M. on December 9 until 7:30 A. M. December 10 about 6,000 tons of water were removed by the use of these twenty pumps.

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A NEW device has been put on the market by A. E. Hoermann, 261 Broadway, New York City, dealer in pneumatic and other machinery, which promises to have a good field.

The tool is supposed to be used in connection with an ordinary air or electric drill and to perform the work of screwing up, tightening and also unscrewing nuts.

Without doubt the largest field for this tool will be the railroad market. On some of the wooden freight cars, for instance, there are as many as 600 nuts of one kind, and needless to say the time consumed in tightening these by hand with a wrench is considerable.

The idea of using a tool of this kind is not a new one, but so far no satisfactory one has been placed on the market. Tests have been made by using solid chucks in drills, but the sudden jar caused at the moment the nut was drawn up on the surface soon put the drill out of commission.

The tool, built according to A. E. Hoermann's ideas, has an adjustable friction arrangement, which enables the

operator to regulate the tension for the largest nut to be tightened.

The nuts to be tightened are placed on the bolts with about one and one-half turn, the tool will run them up to the surface and tighten them, then the friction arrangement will take up the jar and allow the drill to continue rotating while the wrench chuck itself remains stationary.

This tool has been thoroughly tested at the shops of the Central Railroad of New Jersey, at Elizabethport, and found to work remarkably well. It has been found, however, that a three-quarter-inch nut is the maximum capacity, not of the tool itself, but of the drill, as the largest drill is not strong enough to develop power to tighten a one-inch nut sufficiently tight.

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An article in *The Railway and Engineering Review* gives the following description of the piping system used in the Sedalia shops of the Missouri Pacific Railway:

"The compressed air is carried in three independent lines from the power house under ground to the various buildings, and the freight car repair yard. The pipe used for this purpose is Universal cast-iron pipe, manufactured by the Central Foundry Co., of New York. This pipe has special machined ends, with lugs cast at the ends so that the pipe may be laid in sections and bolted together. The pipe is laid in the ground with no further protection than a coat of coal tar. The line for the freight car repair yard leaves the power house at the west end and extends in a straight line about 1,700 feet west. A branch line leaving the main runs to the store house for the operation of air hoists; another line supplies the freight car repair shed and the scrap platform, and at 200 feet intervals branch lines are taken from the main and run north across all the tracks of the freight car repair yard. The pipe of both main and branches is cast with bosses at 3 feet intervals, so that the line may be drilled and tapped, and a riser brought to the surface of the ground at any one of these bosses. At the end of each of the branch lines is a cast-iron drip-pot, provided with a three-quarter inch blow-out pipe and control valve at the surface of the ground. The line is pitched to drain to the drip-pot and a cast-iron roadway box with a cover plate bearing the inscription "Drip Pot" marks the location of each one of the 19 bleeders on the underground air piping. In addition to these drip-pots, which have a capacity of 15 or 20 gallons each, there are five steel reservoirs or pressure accumulators, which still further collect condensation and provide reserve capacity at points of greatest demand for air."

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At the close of 1904 there were 156,456,370 feet or 29,632 miles of mains for the transportation of natural gas in the United States, which varied from 2 to 36 inches in diameter. The greater portion of the pipe in use is 8 and 10 inches in diameter. The line pressure on the smaller pipe up to 10 inches runs as high as 400 pounds; for 20 inches diameter the pressure is usually less than 300 pounds to the square inch. Originally the gas pressure in the wells was sufficient to transport large quantities in ordinary-sized pipes to points of consumption many miles distant. In many instances during recent years the gradual falling off of the natural pressure has made it necessary to instal powerful pumping or compressing machinery.

By far the most economical results have been secured by large gas-engine compressors, in which natural gas is exploded instead of steam being used, and in which 9 cubic feet have developed 1 horse-power per hour, while working at about 1,500 horse-power capacity. Thirty cubic feet of gas have been compressed from 0 to 270 pounds pressure to the square inch by the consumption of 1 cubic foot of natural gas, whereas double that amount of gas is consumed under boilers supplying double-expansion condensing steam engines to do the same work.

In recent years there has developed a general extension of many of the large lines, supplying central western Pennsylvania and northeastern Ohio, into the deep and prolific fields of southwestern Pennsylvania and western West Virginia, which are now operated in a most economical way, enforced by the dearly learned lessons of the reckless exhaustion of the original fields of Pennsylvania, Ohio and Indiana. Much larger lines have been recently constructed, and the increase in diameter has added largely to their capacity and has increased the quan-

tity of gas delivered at points of consumption.—*Engineering and Mining Journal.*

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A CORRESPONDENT of the *Milwaukee Sentinel* writes from Ironwood, Mich., as follows:

"An interesting sight at Corrigan, McKinney & Co.'s Great Western mine at Crystal Falls, Menominee range, is a combination pump operated by compressed air. The history of this unique device dates back several years, when a collapse of the surface ground, caused by the caving system of mining, carried down stopes and threatened the life of the shaft. A considerable portion of the cave, approaching 200 feet in depth, was thereupon filled with sand and the stability of the shaft was restored. It was necessary to instal pumps in order to prevent the flooding of the mine, but instead of pumping the water direct by steam, a method which on account of the sand would be destructive to the ordinary type of pumping machinery, a small air compressing plant and vacuum pump were put in and two cylindrical tanks, each of a capacity of 2,000 gallons, were installed in the bottom of the pit. Each tank is equipped with a water inlet, discharge pipe, and check valve at the bottom, and a compressed air and vacuum pipe at the top. The tanks, connected, occupy space in a reservoir which is the receiving vat for all the surface water.

"While the water is forced into one tank by compressed air, the same movement of the pump drives the air out of the other tank, creating a perfect vacuum and forcing the water into the pipe which conveys it to the top of the pit and into the sluiceway by which it is carried out of range of the mine. When the tank is emptied of water, the air in it, which is under compression, is automatically withdrawn and taken back into the compressor, losing only a part of its pressure in the process and adding materially to the economy of the device. The two tanks operate in balance, one filling while the other is discharging, and thus a continuous stream of water is maintained. The only parts of the device to which the sand in the water can exert a destructive force is on the check valves, saving greatly in wear and tear."

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THE American Institute of Mining Engineers, which for the two days preceding has been in annual session at Bethlehem, Pa., was on Saturday, February 24th, entertained by the Ingersoll-Rand Company at its works at Phillipsburg, N. J.

A special train brought the visiting party, about 165 in number, from Bethlehem direct to the shop grounds. The guests were received by President W. L. Saunders of the company, assisted by members of the sales and engineering departments both from Phillipsburg and New York. The party was at once photographed on the steps of the office building, and each guest was presented with a leaflet showing a plan of the works.

The guests, divided into parties, were piloted by representatives of the company over the specified route through the shops, inspecting the many classes of machinery built there, both in construction and in operation. After a morning spent in this manner, the visitors were conducted to the dining room on the third floor of the office building, where a luncheon was served to about 195 guests and company representatives.

After an hour of fellowship and good cheer, a second photograph was taken and the company was called to order by Mr. De Saules, acting chairman of the entertainment committee. After a few happy remarks of appreciation of the day's entertainment, the chairman called upon President Saunders of the Ingersoll-Rand Company, who responded, calling attention, among other things, to the fact that the Institute was not up-to-date in that it still clung to the hand hammers on its emblematic seal, which should properly be replaced by a machine drill as more typical of modern engineering progress.

Mr. Jasper Rand, vice-president of the company, was then called upon and dwelt in his remarks upon the close and intimate relations in the new company which had taken the place of the keen rivalry between the former Ingersoll and Rand Companies. The next speaker was Mr. Edmund C. Pechin, one of the charter members of the Institute, who gave some historical reminiscences. He was followed by Dr. Raymond, secretary of the Institute, since its foundation. The informal program closed with a song by Prof. Franklin, of Lehigh University, an Institute member, which told of the intimate relations of "Sandy and his Mule" in the days before machinery was introduced in the coal mine. The party then adjourned, returning by their special train.

A number of ladies, guests of the Institute, were in attendance, lending to the occasion the charm of their presence.

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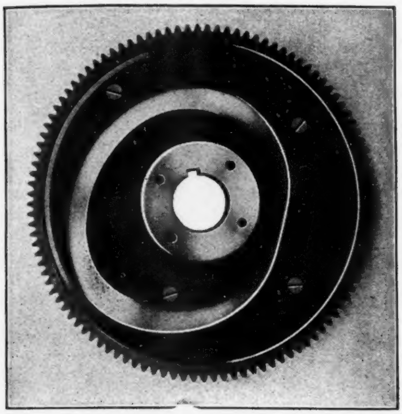
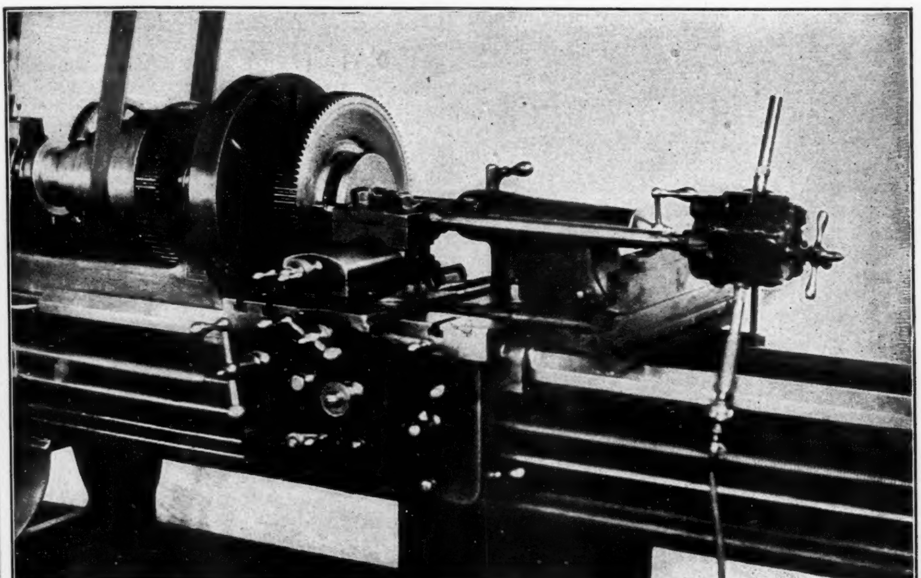
THE accompanying illustrations show an irregular box cam and a novel method introduced for cutting it. It will be observed that at one point the groove is located very close to the centre of the gear, while it runs close to the rim at another point; hence it could not be cut with a single tool, it being absolutely necessary that the groove of the cam be parallel.



Not having a machine particularly adapted to the work an emergency method was devised consisting of a block mounted on slide of engine lathe in which a milling cutter was free to revolve. A master cam was attached to mandrel with gear in proper relation to it; as the lathe revolved slowly (slide rest having previously been disconnected from cross-feed screw) the milling cutter arbor was kept in contact with master cam by means of weight fastened to end of cross-feed slide. The cutter arbor in contact with master cam was the same diameter as the milling cutter used; this insured an absolutely parallel slot. A "Little Giant" air drill was used for driving the cutter. The arrangement proved very satisfactory as the speed could be controlled to a nicety; it made the fixture at minimum cost; no cumbersome belts or other complicated mechanism being necessary, and the set-up required no more time than an ordinary chucking job. We are indebted to Mr. Henry J. Kimman, superintendent of the Cleveland plant of the Chicago Pneumatic Tool Company, for the illustrations and explanation covering the work.



## COMMUNICATIONS.

**Under this heading will be published inquiries addressed to the Editor of COMPRESSED AIR. We wish to encourage our readers in the practice of making inquiries and expressing opinions.**

**We request that the rules governing such correspondence will be observed, viz.: all communications should be written on one side of the paper only; they should be short and to the point.**

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*To the Editor of* COMPRESSED AIR:

We have a problem in compressed air pumping to be solved which we herewith submit to you.

We have a well 720 feet deep cased with steel pipe 7⅝ inches in diameter to 470 feet; there is an intermission of 20 feet in which there is no casing, then 220 feet of 5⅝-inch casing. At 470 feet natural gas and oil were obtained. At 720 feet natural gas and salt water were obtained. The water rises to the top of the well, carrying with it oil and gas, which bubbles over the top of the well presumably two to four gallons per minute.

Many attempts within the past three months have been made to pump the well dry by means of a deep well pump operated by compressed air, the air compressor giving 70 cubic feet at 80 pounds pressure, and the results achieved have been discouraging. Our purpose is to pump the well dry so that we may secure the gas which comes in about 700 to 710 feet which is just above the salt water.

Yours very truly,

BRYAN OBEAR.

Provided the well yields more water than the deep well pump can handle, the air lift system is the only method to employ promising any possibility of lowering the water to a point where the water seal will be broken, and the gas under a certain amount of pressure, liberated to escape with the water pumped. The question of just how far the water level in the well will have to be lowered (the static head reduced) in order to release the gas being unknown, depending upon the gas pressure, the necessary capacity of the compressor and pressure of air cannot be determined at the time.

Considering the air lift system, it will not be practicable to pump the water down to the very bottom of the well—"to pump it dry"—as you state, because this method of pumping depends for its proper operation upon a certain amount of submergence for the piping, or, in other words, the air and discharge pipe must be under water for a certain proportion of their total length in order to give satisfactory results.

We believe, however, using the air lift system, you can lower the water level and reduce the water pressure now holding the gas to such an extent that by pumping into a tank fitted with baffles, or arranged in such a way as to readily separate the gas from the water, that the desired results can be obtained.

It is hardly possible that this can be done with a compressor the size mentioned, but it will be possible to get a line on the requirements by fitting up and testing with this machine, and it will cost but very little to make such a trial.—EDITOR.

*To the Editor of* COMPRESSED AIR:

With reference to the power for use at the well which is intended to be pumped, I beg to say that the compressor will give about 70 cubic feet of free air per minute at 80 pounds pressure.

The air compressor is a hydraulic compressor, being driven by a steam pump to furnish an artificial head of water which is alternately driven into tanks compressing the air above it at the rate of about a revolution every 45 seconds. It is possible to secure 80 feet per minute at a 100 pounds pressure. I enclose you a small blue print of the invention which will give an idea of my hydraulic air compressor.

With reference to the depth which we have been able to pump the well, beg to say that the water has been lowered in it by means of a deep well pump to about 550 feet. At this point the pump stopped working and refused to take the air. We have just taken the pump out and find that the valves were glotted to such an extent with oil that they did not perform their proper functions. During the pumping of the well there was no accurate test of efficiency made, but we estimate that we pumped one gallon of water to two cubic feet of free air. At the next attempt to pump the well I shall pump the water into a tank and measure it accurately and keep a record of the air compressor.

Yours very truly,

BRYAN OBEAR.

# INDEX.





## U.S. PATENTS GRANTED JAN., 1906.

Specially prepared for COMPRESSED AIR.

808,623. CARRIER FOR PNEUMATIC TUBES. Birney C. Batcheller, Philadelphia, Pa., assignor to The Pearsall Pneumatic Tube and Power Company, New York, N. Y., a Corporation of New York. Filed Mar. 21, 1905. Serial No. 251,210.

808,638. FLOATING CAISSON FOR REPAIRING FLOATING DRY-DOCK BOTTOMS. Andrew C. Cunningham, Annapolis, Md. Filed Mar. 10, 1905. Serial No. 249,398.

An under-water-floating caisson for repairing the bottoms of floating dry-docks, the same comprising an open-topped working chamber having a bottom and closed on all sides, said chamber being adapted to be entirely submerged and floated under the dock; and provided with means for exhausting the water therefrom, and having a shaft extending above water to permit workmen to enter the chamber after it is in place.

808,680. AIR-GUN. David F. Polley, Plymouth, Mich., assignor to Markham Air Rifle Company, Plymouth, Mich. Filed Dec. 19, 1904. Serial No. 237,572.

808,703. LOAD-BRAKE APPARATUS. Walter V. Turner, Wilkinsburg, Pa., assignor to Westinghouse Air Brake Company, Wilmerding, Pa., a Corporation of Pennsylvania. Filed Apr. 1, 1904. Serial No. 201,167.

808,711. COMBINED AIR-BRAKE-SETTING DEVICE AND ALARM. Robert E. Adreon and Francis M. Stambaugh, St. Louis, Mo.; said Stambough assignor to said Adreon. Filed Apr. 14, 1905. Serial No. 255,664.

808,716. BRAKE SYSTEM. Charles E. Barry, Schenectady, N. Y., assignor to General Electric Company, a Corporation of New York. Filed Aug. 13, 1903. Serial No. 169,349.

808,728. FLOATING CAISSON. Andrew C. Cunningham, Washington, D. C. Filed Sept. 20, 1905. Serial No. 279,347.

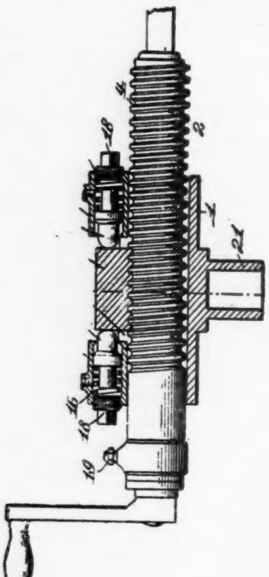
An under-water floating caisson for the purposes described, comprising an open-topped working chamber having a bottom and closed on all sides, said chamber being adapted to be entirely submerged and to be floated under the structure to be repaired, means for gaining access to the chamber from above the water after it is in place and for exhausting the water therefrom, and

floats exterior to the chamber for suspending or maintaining it at the desired depth of submergence while being moved into place.

808,768. COMBINED DOOR, LOCK, AND INDICATOR FOR CARRIERS USED IN PNEUMATIC TUBES. Patrick J. Madigan, Chicago, Ill. Filed Feb. 25, 1905. Serial No. 247,309.

808,811. AUTOMATIC PRESSURE-RETAINER. James W. Brown, Minneapolis, Minn. Filed Aug. 27, 1904. Serial No. 222,386.

808,837. ROCK AND COAL DRILL. Lester C. Hardsocg, Ottumwa, Iowa. Filed May 17, 1905. Serial No. 260,786.

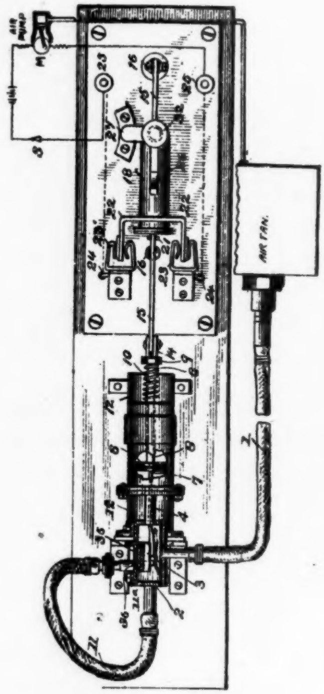


A housing, a hammer mounted therein and provided with exterior screw-threads, a slidably-mounted hinged block having screw-threaded engagement with said hammer, spring-pressed plungers located at opposite ends of said block for cushioning the movements of the same, means for adjusting the tension of the springs of said plungers, and means for turning said hammer.

808,974. PROTECTOR FOR PNEUMATIC TIRES. Henri David, Paris, France. Filed Jan. 20, 1904. Serial No. 189,856.

808,903. AUTOPNEUMATIC MUSICAL INSTRUMENT. Walter Crippen, New York, N. Y., assignor to Wilcox & White Company, Meriden, Conn., a Corporation of Connecticut. Filed June 5, 1905. Serial No. 263,710.

809,083. AUTOMATIC ELECTRICAL CONTROLLER FOR AIR-COMPRESSORS. Fred D. Ackerman, Kendallville, Ind. Filed May 17, 1905. Serial No. 260,865.



The improved automatic controller for the purpose specified, comprising a valve-chest and valve, an air-cylinder and a slidable piston contained therein and connected with said valve, a spring for opposing the movement of the piston in one direction, a second air-cylinder having a spring-pressed piston, an air conductor connecting such second cylinder with the valve-chest, a slidable bar connected with the last-named piston, a casing supplied to and slidable on said bar, a spring contained within the said casing and applying pressure between it and the bar, an electrical switch one member of which is connected with the casing, and spring locking and release mechanism connected with the casing for holding it nor-

mally immovable and the switch normally closed until a certain predetermined air-pressure is accumulated.

809,206. MINING-MACHINE. Daniel R. Murphy, Scenery Hill, Pa., assignor to Joseph A. Jeffrey, Columbus, Ohio. Filed Feb. 20, 1904. Serial No. 194,586.

The combination of a motor platform or carrier, a cutter-frame having a central thrust-rail adjustable relatively to the motor-carrier and having its rear end loosely braced, and a connecting and bracing bar projecting forward from the motor-carrier and adjustably clamped to the said centre rail of the cutter-carriage, substantially as set forth.

810,061. PNEUMATIC-CUSHION WHEEL. Charles A. Lee, Kansas City, Kans., assignor of one-half to Albert J. Holzmark, Kansas City, Kans. Filed June 28, 1905. Serial No. 267,471.

810,298. AIR-LOCK FOR CAISSONS. William McIlvrid, Jersey City, N. J., assignor to The Cockburn Barrow and Machine Company, Jersey City, N. J., a Corporation of New Jersey. Filed June 22, 1905. Serial No. 266,488.

A casing having a stationary top portion extending partially over the upper end thereof, said casing having an opening in its side extending lengthwise thereof to permit of removing and replacing the bucket, and means for simultaneously closing said opening and that part of the top of the casing not covered by said stationary portion.

810,316. EMERGENCY RAIL-BRAKE. Powell O. Adams, Cameron, Tex. Filed July 6, 1903. Serial No. 164,194.

809,409. PNEUMATIC TIRE. Pardon W. Tillinghast, Edgewood, R. I. Filed Mar. 5, 1903. Serial No. 146,396.

809,530. PNEUMATIC-TIRE COVER. William A. Sankey, Sutton, England, assignor to Frank Reddaway, Manchester, England. Filed Jan. 24, 1905. Serial No. 242,544.

809,633. SUBMARINE MINING. Edgar F. Scheibe, New York, N. Y., assignor, by direct and mesne assignments, to Submarine Gold Mining Company, a Corporation of New York. Filed Mar. 1, 1905. Serial No. 247,904.

In combination, a support arranged to rest on the bottom, and a caisson supported thereby and adapted to extend to the bottom to permit expulsion of the water therefrom within the area of the caisson.

809,707. PNEUMATIC TRAIN-CONTROL SYSTEM. John B. Linn, Schenectady, N. Y., assignor to General Electric Company, a Corporation of New York. Filed Oct. 16, 1902. Serial No. 127,464.

809,774. SYSTEM OF CONTROL. Frank E. Case, Schenectady, N. Y., assignor to General Electric Company, a Corporation of New York. Filed June 12, 1905. Serial No. 264,763.

809,908. SYSTEM OF CONTROL. Frank E. Case, Schenectady, N. Y., assignor to General Electric Company, a Corporation of New York. Filed Oct. 1, 1904. Serial No. 226,755.

809,919. PNEUMATIC MOTOR. Lawrence U. Jobes, Cincinnati, Ohio. Filed Dec. 5, 1904. Serial No. 235,545.

The combination with a bellows, of a bellows-passage communicating therewith, a valve-passage leading from the bellows-passage to the exhaust-passage; an exhaust-passage communicating with the valve-passage by way of a port; a valve controlling said port; a pneumatic placed in said exhaust-passage operating said valve; a pneumatic-passage leading from said pneumatic in the exhaust-passage to the outer air with a pneumatic-port at the outer end thereof; an air-passage communicating with said bellows, having an air-port at the outer end thereof, the pneumatic-port and air-port being controlled by a single slide-valve.

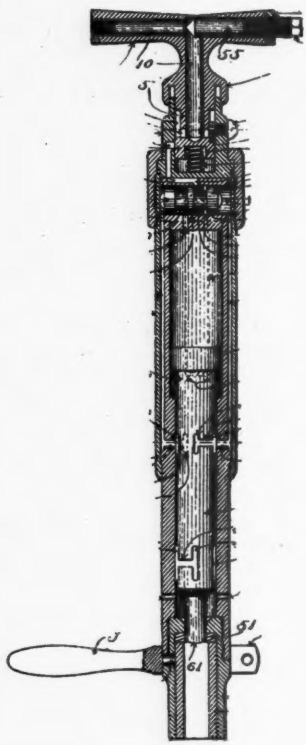
810,428. TUNNEL-SHIELD. Charles L. Parmelee, Orange, N. J., and Edward G. Williams, Washington, D. C. Filed Jan. 3, 1905. Serial No. 239,299.

810,450. APPARATUS FOR TREATING AIR. Frederick White, Boston, Mass., assignor to Regenerated Cold Air Company, Portland, Me., a Corporation of Maine. Filed Sept. 4, 1903. Renewed Nov. 11, 1905. Serial No. 286,790.

810,451. APPARATUS FOR TREATING AIR. Frederick White, South Boston, Mass., assignor to Regenerated Cold Air Company, Kittery, Me., a Corporation of Maine. Filed Aug. 8, 1904. Renewed Nov. 11, 1905. Serial No. 286,791.

810,514. FLUID-PRESSURE BRAKE. Walter V. Turner, Wilkinsburg, and John S. Custer, Pittsburg, Pa., assignors to The Westinghouse Air Brake Company, Pittsburg, Pa., a Corporation of Pennsylvania. Filed Apr. 27, 1904. Serial No. 205,116.

810,603. PNEUMATIC TOOL. William E. Badger, Quincy, Mass. Filed July 20, 1904. Serial No. 217,364.

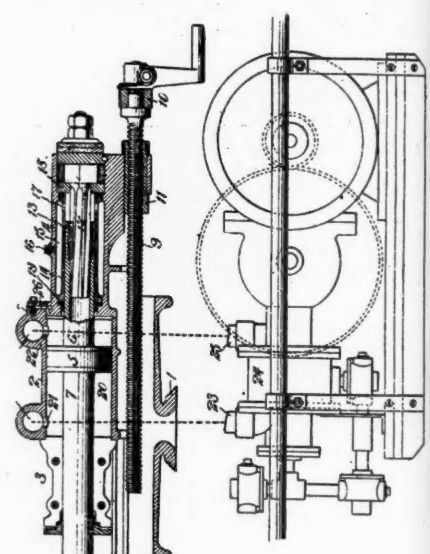


A pneumatic impact-tool, including a cylinder and piston, a valve-casing having a valve therein for controlling the movements of the piston, a head secured to said cylinder and held in fixed relation thereto and to said valve-casing, a handle or support upon which said head is rotatably mounted, said head having a passage for motive fluid which is alternately placed in communication with opposite ends of the cylinder, a passage in said handle which may be connected with a source of fluid-pressure supply and is adapted to be placed in communication with said passage in the head, and means to rotate said cylinder with relation to said handle and source of fluid-pressure supply.

810,609. ELECTROFLUID PRESSURE MECHANISM FOR OPERATING RAILROAD APPLIANCES. Walter J. Bell, Los Angeles, Cal., assignor of one-half to Leon F. Moss, Los Angeles, Cal. Filed Dec. 1, 1903. Serial No. 183,355.

810,624. PRESSURE-CONTROLLING DEVICE FOR THE AIR SPRINGS OF PRINTING-PRESSES. Charles P. Cottrell, Westerly, R. I., assignor to C. B. Cottrell & Sons Company, New York, N. Y., a Corporation of New Jersey. Filed Apr. 4, 1905. Serial No. 253,787.

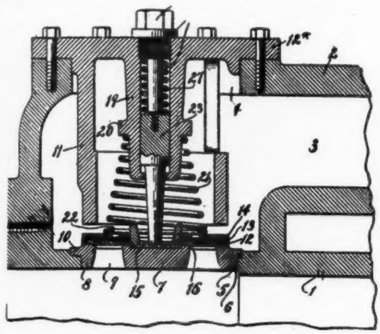
810,639. ROCK-DRILL. Arthur H. Gibson, Easton, Pa., assignor to The Ingersoll-Sergeant Drill Company, New York, N. Y., a Corporation of West Virginia. Filed June 9, 1905. Serial No. 264,433.



The combination with a rock-drill cylinder having a rear extension, of a piston having a hollow tail-rod extended into and entirely inclosed by said rear extension, a sleeve surround-

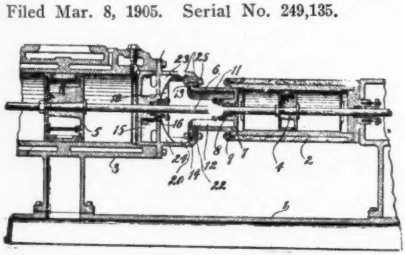
ing the tail-rod forming an annular chamber within the rear extension, a rifle-bar telescoping within the tail-rod and a port leading from the annular chamber to the air-space within the sleeve and tail-rod.

810,672. DISCHARGE-VALVE FOR FLUID-COMPRESSORS. William Prellwitz, Easton, Pa., assignor to The Ingersoll-Sergeant Drill Company, New York, N. Y., a Corporation of West Virginia. Filed Apr. 1, 1905. Serial No. 253,247.



A valve-seat, a valve, its spring for yieldingly holding the valve closed, a removable valve-guide engaged with the seat and means for yieldingly holding the valve guide and seat in position.

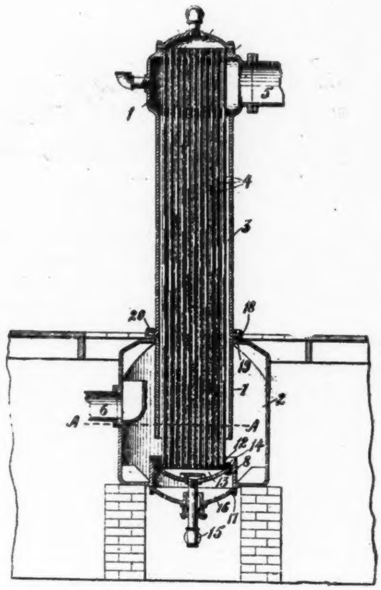
810,671. SEPARABLE HEAD-COUPLING. William Prellwitz, Easton, Pa., assignor to The Ingersoll-Sergeant Drill Company, New York, N. Y., a Corporation of West Virginia. Filed Mar. 8, 1905. Serial No. 249,135.



A separable head-coupling for alined cylinders comprising two cylinder-heads having shanks removably secured together, one of the heads when released being arranged to permit its shank to telescope within the shank of the other head.

810,725. BELL-RINGER. Richard M. Crosby, Tacoma, Wash. Filed Aug. 2, 1905. Serial No. 272,298.

810,670. COOLER FOR COMPRESSORS. William Prellwitz, Easton, Pa., assignor to The Ingersoll-Sergeant Drill Company, New York, N. Y., a Corporation of West Virginia. Filed Apr. 14, 1903. Serial No. 152,582.

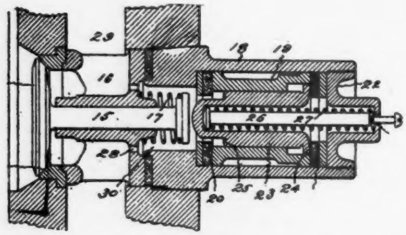


A cooler for compressors comprising a stack of cooling-tubes, a casing therefor having an upper fluid-inlet head, a lower fluid-discharge head, and a connecting body portion extended into the interior of the fluid-discharge head about the portion of the stack of tubes extended therein and removably secured to the discharge-head whereby the air is confined to the stack of tubes within the discharge-chamber and the cooling-tubes may be removed from the discharge-head by a comparatively short upward movement of the body portion.

810,759. VALVE FOR COMPOUND AIR-COMPRESSORS. Ebenezer Hill, Norwalk, Conn. Filed Jan. 24, 1905. Serial No. 242,498.

A valve for the intake-passage of an air-compressor having a valve-disk adapted to open and close the passage, a plunger movable toward and from the disk, said plunger having a surface at its inner end that is exposed to the air-pressure in the intake-passage when the plunger is at its inner limit, and a larger surface back from its

inner end that is exposed to the air-pressure in the intake-passage when the plunger is moved from its inner limit, whereby the pressure against the smaller surface will tend to move the plunger away from the disk and pressure against

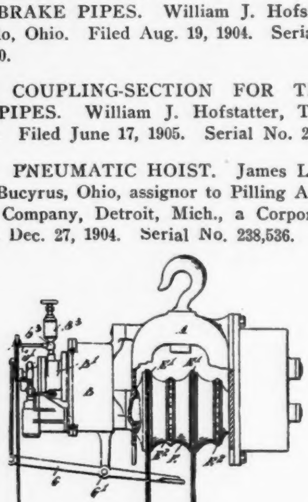


the larger surface will tend to hold the plunger away from the disk, and a spring arranged to force the plunger against the air-pressure when the pressure drops below normal and cause the plunger to interfere with the ordinary action of the disk.

810,862. SAFETY COUPLING-SECTION FOR AIR-BRAKE PIPES. William J. Hofstatter, Toledo, Ohio. Filed Aug. 19, 1904. Serial No. 221,370.

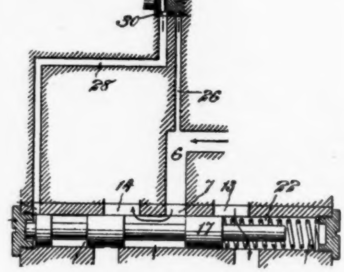
810,863. COUPLING-SECTION FOR TRAIN AIR-PIPES. William J. Hofstatter, Toledo, Ohio. Filed June 17, 1905. Serial No. 265,697.

810,884. PNEUMATIC HOIST. James L. Pilling, Bucyrus, Ohio, assignor to Pilling Air Engine Company, Detroit, Mich., a Corporation. Filed Dec. 27, 1904. Serial No. 238,536.



A hoisting mechanism, including the combination of a rotatable horizontal shaft, a plurality of sprockets thereon, a loop-chain having its two strands passing over two sprockets, a motor to rotate the shaft, a throttle-valve to control said motor, and means adapted to be actuated manually or by contact with another portion of the mechanism whereby said valve may be actuated.

810,890. REVERSING MECHANISM FOR ROTARY MOTORS. Albert H. Taylor, Easton, Pa., assignor to The Ingersoll-Sergeant Drill Company, New York, N. Y., a Corporation of West Virginia. Filed July 14, 1905. Serial No. 269,607.



A motor, means for supplying motive fluid thereto, a reversing-valve, passages leading from the motive-fluid supply and the opposite sides of the motor piston-chamber to the face of the valve, a port in the valve for alternately opening communication from the fluid-supply passage to the one or the other of the motor piston-supply passages, exhaust-ports opening to the face of the said valve and ports in the valve for alternately opening communication from the one or the other of the motor piston-supply passages to the one or the other of the said exhaust-passages, a spring for moving the reversing-valve in one direction and means for controlling the supply of motive fluid for moving the valve in the opposite direction.

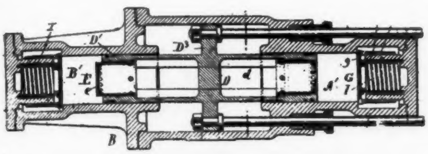
811,064. AIR-BRAKE. Edwin T. Hughes, Gladstone, Colo., assignor of one-fourth to Vincent J. O'Brien, one-fourth to Willis Z. Kinney, and one-fourth to James M. Rogers, Silverton, Colo. Filed Oct. 9, 1905. Serial No. 281,912.

811,109. PNEUMATIC TIRE. Friedrich Veith, Hochst-in-the-Odenwald, Germany. Filed Apr. 29, 1905. Serial No. 257,971.

811,110. APPARATUS FOR PUMPING LIQUID. George L. Waterhouse, New York, N. Y. Filed June 13, 1905. Serial No. 264,984.

An apparatus comprising a plurality of receptacles provided with liquid induction and eduction means, means for controlling the flow of steam to each receptacle to accord with the rise and fall of liquid therein, means for discharging heated fluid from the lower interior of one receptacle into the upper interior of another receptacle upon a charge of liquid therein before steam is admitted thereto, and means for causing condensation of steam in said receptacles after liquid has been discharged therefrom.

811,181. GAS-COMPRESSOR. Louis Sterne, London, England. Filed Aug. 18, 1904. Serial No. 221,279.



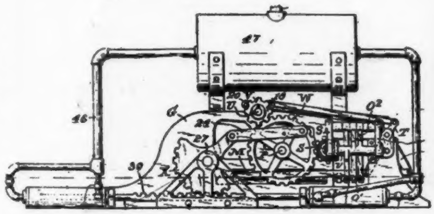
A double-acting gas-compressor, composed of two tandem single-acting single-stage compressor-cylinders, each having a delivery-valve at its outer end, a gas-tight casing of larger diameter connecting the adjacent inner ends of the cylinders, a return-gas inlet to the casing, two connected hollow trunk-pistons working in the respective cylinders, each such piston carrying a suction-valve and apertured to afford communication between the casing and the valve, radial arms on the pistons, and piston-rods connected to the arms and passing through lateral stuffing-boxes at one end of the casing, whereby the cylinders, pistons and piston-rods are kept at a moderate uniform low temperature.

811,296. PNEUMATIC SPRING FOR VEHICLES. Matthew M. Howland, Providence, R. I., assignor of one-half to William W. Dunnell, Providence, R. I. Filed May 11, 1905. Serial No. 259,969.

811,330. PRESSURE AND SUCTION DEVICE. Eugen Roth, Schoneberg, near Berlin, Germany. Filed May 6, 1904. Serial No. 206,759.

An air-tight chamber having an elastic flexible wall, and a series of devices conforming to and movable with said wall, to prevent disruption thereof; with levers contacting said devices, a reciprocity rod, and an operative connection between the rod and levers.

811,378. AIR-COMPRESSING ENGINE. Hobbs E. Clifford, Lafontaine, Ind. Filed Apr. 17, 1905. Serial No. 255,926.



An engine including a support having a chamber therein, a hollow cylindrical abutment secured fixedly at an end thereof to the support in communication with the chamber thereof and provided at the opposite end thereof with a valve-seat, a valve operative on the valve-seat and having an operating-rod extending through the chamber of the support, a hollow cylinder movable on the abutment, a rotative shaft having a crank, operative connections between the hollow cylinder and the crank of the shaft, and connections between the rotative shaft and the operating-rod of the valve.

811,400. PNEUMATIC BELL RINGER. John Howe, St. Joseph, Mo. Filed May 6, 1904. Serial No. 206,772.

The combination of a bell-crank for an alarm-bell, a cylinder provided with inlet and exhaust ports in the wall at one side thereof only, a reciprocating piston mounted therein and provided with inlet and exhaust passages arranged to be connected and disconnected alternately with the inlet and exhaust ports in the cylinder, means for keeping the reciprocating piston seated on the port side of the cylinder-wall, and means for connecting the reciprocating piston with the bell-crank.

811,507. AUTOMATIC AIR-BRAKE APPARATUS FOR AUTOMOBILES. Frederick Kalisch, St. Louis, Mo., assignor of one-eighth to Geo. D. Kluegel, St. Louis, Mo. Filed Mar. 24, 1905. Serial No. 251,903.

811,525. AIR-BRAKE. William L. Stiarwalt and William H. Cool, Bellefontaine, Ohio. Filed June 22, 1905. Serial No. 266,533.

811,532. PNEUMATIC STACKER. Henry J. Daws, Hopkins, Minn., assignor to Severin N. Sorensen, Minneapolis, Minn. Filed Feb. 6, 1904. Serial No. 192,293.

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